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VAPOR GENERATOR FEED PUMP FOR RANKINE CYCLE AUTOMOTIVE PROPULSION SYSTEM (CHANDLER EVANS)

CHANDLER-EVANS

PREPARED FOR
ENVIRONMENTAL PROTECTION AGENCY

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FOR RANKINE CYCLE
AUTOMOTIVE PROPULSION SYSTEM
(CHANDLER EVANS)

CHANDLER EVANS REPORT NO. R-679-5

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A project was undertaken to conduct comprehensive design studies pursuant to the selection of conceptual models of vapor generator feed pumps that will satisfy the performance requirements of each of three Rankine cycle automotive power systems currently under development. In pursuing the objective of providing variable output pumps for these applications, investigations were conducted of two selected courses: (1) fixed displacement pumps with variable speed drives, and (2) variable displacement pumps. The report presents a detailed summary of the project, describes the technical results, and gives conclusions.

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(CHANDLER EVANS)

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PREFACE

This is the final report submitted by Chandler Evans under the terms of Contract Number 68-01-0430, issued by the Environmental Protection Agency of the United States Government.

As specified in Exhibit A of the contract, Chandler Evans was to conduct comprehensive design studies pursuant to the selection of conceptual models of vapor generator feed pumps that will satisfy the performance requirements of each of three Rankine cycle automotive power systems currently under development.

In pursuing the objective of providing variable output pumps for these applications, Chandler Evans conducted investigations of two selected courses toward achieving this goal:

1. Fixed displacement pumps with variable speed drives.

2. Variable displacement pumps.

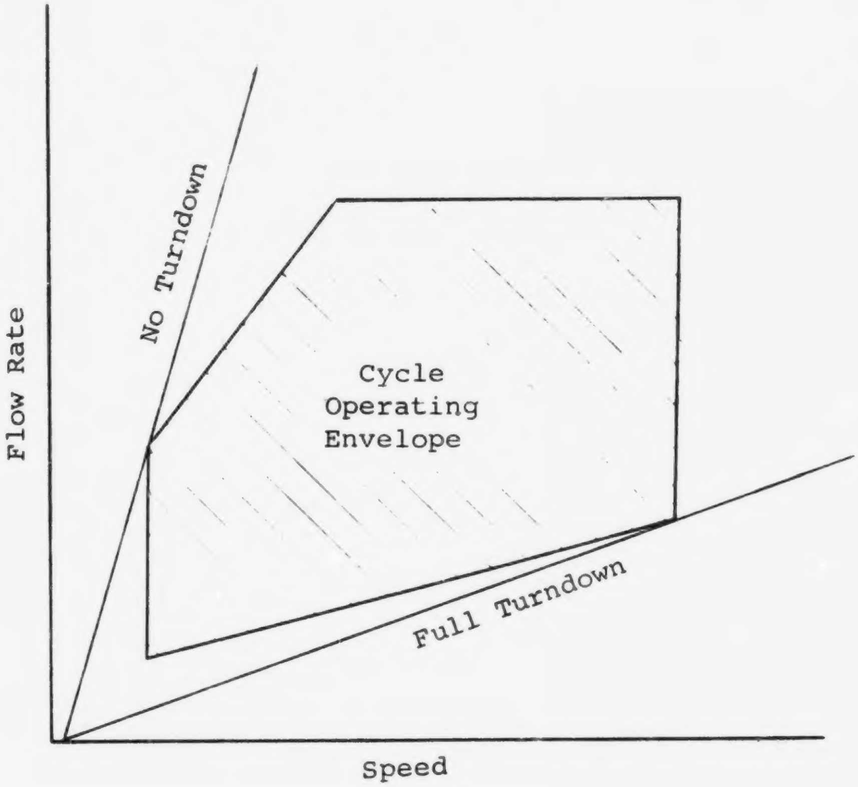
1. INTRODUCTION

The Environmental Protection Agency is working with three companies toward the development of Rankine cycle engines for automotive applications. As summarized below, each of these companies is exploring a different Rankine cycle concept utilizing a different cycle fluid.

Company	Cycle Fluid	Cycle Concept
Aerojet Liquid Rocket Company Sacramento California	AEF-78	Turbine Expander
Steam Engine Systems Watertown Massachusetts	Water	Reciprocating Expander
Thermo Electron Corporation Waltham Massachusetts	Fluorinol 85	Reciprocating Expander

In all cases, a pump is required within the cycle loop to deliver working fluid from the condenser sump to the high pressure boiler. These pumps (called boiler feed pumps) deliver working fluid at pressures from 600 to 1200 psi over very wide flow and speed ranges.

A typical flow-speed operating envelope is illustrated below:



The principal pumping problem associated with the requirements of this application is in maintaining high efficiencies while delivering at all points within the operating envelope. A positive displacement pump is considered the prime choice because of its inherent relatively high efficiency. A typical positive displacement pump has a straight-line flow-speed relationship. The "no turndown" line in the preceding figure represents a positive displacement pump sized to meet the overall requirement of the operating envelope shown. The excess flow must, however, be bypassed as cycle demand varies within the envelope. At "full turndown" as noted on the figure, for instance, we can have turndowns of over 25:1. Thus, a fixed displacement pump must bypass more than 96% of its output at high pressure to meet the extreme lower corner of the envelope. It is apparent, then, that a fixed displacement pump, using a high pressure relief valve to obtain partial flows, will result in inefficient operation.

The next logical step would be to utilize a positive displacement pump with a variable output. This can be done in either of two ways. First, by building a mechanism within the pump, the displacement can be varied at any engine speed. Secondly, a fixed displacement pump can be driven by a speed variator to obtain the wide range of flows through pump speed variation. Again, referring to the preceding figure, the output curve could be rotated from "no turndown" to "full turndown" by varying the input drive speed over the turndown range of 25:1 or whatever the requirement may be. All pump output at high pressure would then go to the boiler, eliminating the need for a bypass valve to achieve the low flows.

This study considers the various options for accomplishing the above noted function. Since the requirements of each of the power plants vary substantially, three separate pumping systems have been studied in order to provide the best overall approach for each application.

2. PROJECT SUMMARY

In the conduct of the parallel program of investigating fixed displacement pumps with variable speed drives and variable displacement pumps, and through contacts with the engine contractors, the following conclusions and positions have been formulated.

2.1 System Considerations

2.1.1 Looking at the general overall operating envelopes of flow and speed for the various boiler feed pumps, the specified turndown ratios approach levels of 26:1.

2.1.2 The data developed for the pumping systems described herein were based on the current published engine requirements. There appears to be a trend toward conservatism in order to insure sufficient turndown margin in the feed pumps. With the narrowing of the requirements that can be expected as the engine programs progress, the real turndown levels should become less severe and more easily obtainable.

2.2 Variable Speed Drives

2.2.1 Conventionally available designs for suitable variable speed drives produce speed ratios in the ranges of 6:1 to 9:1. These may possibly be increased somewhat by extending the existing designs.

2.2.2 The high turndown ratios required by the systems can be obtained by combining a variable speed drive with a differential to expand the available turndown ranges of the drives, while paying a penalty in power consumption, cost. size and weight.

2.3 Positive Displacement Pump

2.3.1 Gear and vane pumps are applicable devices from the standpoints of technology and life capability which have been developed over the years on turbine fuels.

2.3.2 Piston-type pumps have been developed and are available for use in hydraulic fluids, but their dependence on hydrodynamic concepts for life and wear characteristics, plus their inherently higher cost, place them in a somewhat unfavorable position.

2.3.3 Variable displacement vane pumps have proven in practice their ability to handle low viscosity fluids. Cost and performance tradeoffs will establish their rank in priority. Variable displacement piston pumps may warrant consideration if some of the mechanical problems can be overcome.

2.4 System Approach

Based on the preceding points, our study points toward a system with either a gear or vane type pump in combination with a variable speed drive.

Furthermore, since the high flows are needed only during acceleration transients and at very high speed cruise conditions, a staged pump configuration was investigated. By this technique, the basic pump characteristic follows a much lower capacity limit through most of the operating range, and only at the high demand condition will the high capacity function be switched on. By so doing, the turndown range of the pump and variable speed drive can be narrowed for more efficient operation. This may be accomplished by providing a supplemental pump to be called in on demand or by running a two-stage pump in series normally, to be switched to parallel operation on demand. At the present time, Chandler Evans favors the latter method. This is described in more detail on Page 26.

In keeping the pump-drive package within reasonable performance limits, the expander low-speed, high flow and high-speed, low flow corners of the operating envelope may become inaccessible. In order to achieve performance at these extremes, an accumulator may be used at the high flow condition and a bypass valve at low flow.

2.5 Pump Performance Assessment

Current performance predictions are based on available test data from Chandler Evans' and other manufacturers' pumps running with aviation gasolines and kerosenes or their equivalent calibrating fluids. Any real assessment of performance, including overall performance with the variable speed drive system. must be determined from running these pumps on the actual process fluids.

A Chandler Evans pump has been tested with Fluorinol 85, perchloroethylene (a substitute for AEF-78), and AEF-78 in our Laboratory. Additional testing of this pump is planned with the AEF-78 fluid. Gear and bearing tests have been conducted to ascertain the general wear characteristics of these components in Fluorinol 85, AEF-78, perchloroethylene and water.

A variable displacement vane pump was delivered to Aerojet Liquid Rocket Company to be run in their test loop on AEF-78. ALRC estimates that this testing will commence on or about December 15, 1972.

2.6 Engine Coordination

All of the engine contractors have been visited and the results of our analytical studies and design layouts conveyed to them. Frequent communication has been maintained to coordinate our efforts with their requirements.

2.7 Program Direction

The development of any component in a system intimately affects the entire system. Therefore, integration of the feed pump into the power plant demands investigation of the pump's effect on other components and on the system in general.

No design should be considered as standing on its own merits. Before any pump design is accepted and before any hardware is fabricated, the pump must be satisfactorily coordinated into the whole system.

Chandler Evans submits that this can best be done by the system contractor. We, thus, recommend that before hardware is fabricated, the pump supplier and the engine contractor enter into a period of active cooperation to develop the pump that best fulfills the engine requirements.

3. TECHNICAL RESULTS

3.1 Pumping Requirements

The pumping requirements for the three Rankine cycle systems under study have evolved to some extent over the duration of this contract. The latest flow envelopes to which we addressed our attention are illustrated in Figures 1, 2 and 3.

The Aerojet engine steady-state operating lines for various road grades are superimposed on the overall flow envelope (Figure 3). Although this information is not available for the other systems, it is assumed that they will also follow a similar pattern. This type of information is very valuable in understanding the feed pump's function in the system and has enabled us to better work toward satisfying the engine operating demands.

By applying the linear flow-speed relationship, characteristic of positive displacement pumps at constant pressure, to the corners of the performance envelopes of Figures 1, 2 and 3, the maximum turndown ratios for each system were calculated to be:

TECO: 19.75
SES: 16.45
ALRC: 26

These turndown ratios were calculated by dividing the slopes of the two straight lines through zero which encompass the operating envelope. (Referring to the sketch on Page 2, this is the slope of the "No Turndown" line divided by the slope of the "Full Turndown" line.)

In actual practice, the turndowns will be slightly higher than shown because of volumetric losses, which are primarily a function of pressure.

The specified performance parameters and other pertinent data are listed in Table II.

To provide variable process fluid delivery over the wide turndown ratios, two basic approaches were studied:

1. A fixed displacement pump on a variable speed drive.
2. Variable displacement pump.

In addition to these basic approaches, a fixed displacement pump with bypassing was investigated for the SES engine because its flow rates (and power consumption) are so much lower than the others.

The staging of a fixed displacement pump for series-parallel operation was also examined for efficiency optimization.

All of these concepts will be developed in the ensuing pages and in the Appendices.

3.2 Variable Speed Drives

Table I lists several types of variable speed drives and their salient features. The torque and efficiency characteristics of the currently available types of variable speed drives are illustrated in Figures 4 and 5.

The extremely wide turndown ranges needed by each engine are beyond the limits of the finite-range variable speed drives discovered in our market research and analyzed in this study program.

By employing differentials in conjunction with these drives, very high multiplication in speed ratios can be obtained; thus, extending their ranges. However, this is achieved at the expense of increased power losses and increased unit size, caused by a recirculating power loop within the drive train. The variable speed device must handle this added load plus the torque transmitted to the pump. The net efficiency of the drive train is thereby diminished.

The effect of employing a differential with a variable speed drive is described analytically in Appendix A of this report.

In the interest of minimizing complexity and cost, and maximizing system efficiency, the combination differential/variable speed drive is not recommended unless there is no other recourse.

The necessary speed ranges could be obtained with an infinitely variable speed variator. The hydrostatic transmission is such a device. However, it was eliminated from consideration because of its cost, size, complexity and poor efficiency.

The Lycoming Division of AVCO has an infinitely variable planetary gear drive under development. Its predicted performance characteristics are attractive. It was not considered in the analyses contained herein because of its preliminary nature and the lack of solid definition. In time, however, it may prove to have application in a variable delivery pumping system.

The input to the drive-ratio-varying device must consider the specific requirement of the power plant control system. Ultimately, the position change of a mechanism within the device determines the output. This is usually accomplished by hydraulic pressure acting on a piston or piston/spring combination. The source of the pressure and means for controlling it must be developed into the power plant controlling mechanism by the engine manufacturer.

3.3 Pumps

Only a positive displacement pump can meet the conditions of flow, pressure and efficiency demanded by the Rankine cycle automotive power plants.

The three basic types of positive displacement pumps are described briefly below. The role of inducers in the pumping system is also mentioned.

3.3.1 Piston Pumps have the greatest potential for high efficiencies, but need close clearances to do so. They are somewhat intolerant of contaminant, tend to be bulky and are expensive. A discussion of design considerations for piston pumps is contained in Appendix H.

3.3.2 Vane Pumps are simple but require some precision in manufacturing. They are of moderate cost, tolerate contamination very well and operate at reasonably good efficiencies. Descriptive data and design considerations are contained in Appendix D.

Although several types of vane pumps were investigated, only the balanced-rotor, fixed-clearance type is presented in the trade-off studies reported herein. This configuration offers the best efficiencies because of its inherent low bearing and rotor loading.

3.3.3 Gear Pumps are durable and simple. They have a reasonable tolerance for contamination and are relatively inexpensive. Gear pump efficiencies are within acceptable levels for consideration in this application, although they tend to be lower than those of the piston and vane pumps, especially at low speeds. Descriptive data and design considerations are contained in Appendix C.

3.3.4 Inducers

The dry lift capabilities of Chandler Evans gear and vane pumps are excellent. Therefore, no problems are anticipated at start-up with the minimum liquid heads specified.

However, at operating speeds, the high pressure element of each pump studied must be charged to avoid cavitation. Based on the data contained in Table II, integrated centrifugal inducers were designed for each unit. The development of the design data is explained in Appendix E.

To substantiate the analytical computations, a Chandler Evans Model MFP-150 pump was run at suppressed inlet pressures. The test results shown in Figure 36 indicate that the inducer designs are conservative.

A jet inducer was included in the study of Appendix E for possible application to a bypassing approach for the SES system. A satisfactory inducer of the types shown in Appendix E will preclude the need for any other boost pump in the system.

3.4 Variable Displacement Pumps

Both piston and vane pumps can readily be adapted to variable displacement operation. Such action necessarily will add cost to the pump. Chandler Evans currently has a variable displacement vane pump operating successfully on a development aircraft. Our experience with this pump is reflected in the descriptive information of Appendix F.

Commercially available variable displacement piston pumps have been considered. Prime producers of these devices include Vickers and Sundstrand. Meetings with engineers from these Companies and discussions relating to the use of variable displacement piston pumps with the Rankine cycle fluids, have been discouraging at best. These pumps have been developed to operate with specific hydraulic oils and use the fluid properties to maximize performance. They are dependent on hydrodynamic bearings to minimize friction, and require viscosity/clearance combinations to diminish leakage. Any variable displacement piston pump for use with cycle fluids as they are now known will require an extensive development effort.

The above mentioned Chandler Evans pump has been sent to Aerojet for evaluation with the AEF-78 process fluid.

Any consideration of developing this type of pump must be preceded by a thorough analysis of the test results. The performance and cost aspects must be carefully studied, compared and traded off with respect to the same aspects of the variable speed drive.

3.5 Pump/Drive Approaches

In our effort to develop feed pump systems suitable for the Rankine cycle automotive power plants under study, our attention was primarily focused on performance, especially as reflected in efficiency. However, we were ever cognizant of the need to compete in a large consumer market where both reliability and cost are significant factors.

The basic performance requirements to which we addressed ourselves are summarized in Table II. The flow-speed relationships are illustrated in Figures 1, 2 and 3. These items reflect our latest inputs.

Layout studies were made concurrent with the analytical investigations. These are discussed in a latter section of the report.

The specific approaches for each system are summarized below. All efficiency curves shown for the gear and vane pumps presented herein include the efficiencies of the inducers integrated into each system.

3.5.1 Organic Fluid Reciprocator - TECO

Five distinct variable delivery pump arrangements were analyzed for the TECO engine.

3.5.1.1 Fixed Displacement Gear Pump on a Variable Speed Drive

A gear pump was investigated because of its essential ruggedness and low cost. The particular gear pump analyzed here has a displacement of 0.66 cubic inches per revolution (CIPR).

The plot of Figure 6 is especially descriptive of the operating parameters of a pump on a variable speed drive. The engine requirements are shown in the upper right-hand quadrant. The pump characteristic appears in the upper left-hand quadrant. The lower right-hand quadrant contains the fixed speed ratio line of the expander gearbox.

The lower left-hand quadrant shows the pump speed envelope resulting from the operation of a 9:1 ball-type variable speed drive on drive pad speed to obtain the desired flows.

As indicated in Figure 7, the pump-drive overall efficiency falls short of meeting most of the specification limit points, although it does approach them.

3.5.1.2 Fixed Displacement Vane Pump on a Variable Speed Drive

Figures 8 and 9 show the effect of a 0.6 CIPR vane pump in the system. Although the peak efficiency of the vane pump chosen for this application is about the same as that for the previously described gear pump, its better off-design characteristics bring it closer to meeting the full range of specification limits.

The spool-type variable speed drive was selected to further enhance efficiency. (See Table I.) However, this drive is unable to reduce pump speed sufficiently to pick up the low flow points, as illustrated in Figure 8. Therefore, a bypassing relief valve must be employed to permit the necessary flow reduction at these conditions. The sharp drop off in efficiency seen in Figure 9 on the 1800 rpm line is a result of this bypassing action.

Since relief valves are common to all of our positive displacement pumps for protection, the inclusion of the bypassing function to fulfill the complete performance envelope does not add hardware to the system.

3.5.1.3 Two-Stage Vane Pump on a Variable Speed Drive

To better utilize the full-range efficiency characteristics of the pump, a two-stage arrangement was devised operating on the spool-type variable speed drive. The schematic of Figure 10 illustrates the basic function of this concept.

At low demand, the discharge of one stage is fed to the inlet of the other, splitting the pressure rise between the two elements. They are, thus, operating in series. The displacement of each element in this application is 0.3 CIPR.

When the demand exceeds the capability of this combination. both elements go into parallel operation. doubling the flow capacity. The speed-flow map of Figure 11 produces the improved efficiencies shown in Figure 12. The switching point will be a function of pump capacity, system considerations and the nature of the control mechanism.

Referring to Figure 12, it can be seen that the optimum switching points are at the intersections of the efficiency curves for series and parallel operation at each speed. In this case, the flow-speed relationship at optimum switching is linear. This suggests the suitability of a simple speed actuator. Deadband must be built into the device to avoid oscillation at the switching point.

For more descriptive information about two-stage pump operation, refer to Appendix I.

3.5.1.4 Variable Displacement Vane Pump

To ascertain the effect of a variable displacement vane pump in the Rankine cycle system, Chandler Evans drew upon its experience with a high speed variable displacement vane pump currently going into production for an aircraft application. Running this pump slower allows more flexibility in materials selection with a consequent reduction in both power consumption and cost. To further enhance performance, the proven technique of providing hydrodynamic vane tip bearing was considered. The resultant efficiencies are shown in Figure 13.

A pump of this type will require an extensive development program. The performance gain would have to be weighed against the cost of such a program and the inherently higher unit cost.

3.5.1.5 Variable Displacement Piston Pump

Since TECO currently has a development program underway for a variable displacement piston pump, we have plotted their published efficiencies in Figure 14 for comparison purposes. The power losses of the TECO condenser-mounted boost pump were neglected in determining these efficiencies.

3.5.2 Water Reciprocator - SES

Based on our past experience, vane pumps only were considered for the SES engine because of the material limitations imposed by using water.

Furthermore, because of the lower power demand of the feed pump in the SES system, other than variable delivery pumps are among the five concepts discussed below.

Many of the statements made in the section on TECO pumps (3.5.1) apply here also.

3.5.2.1 Fixed Displacement Vane Pump on a Variable Speed Drive

This arrangement utilizes the more efficient but smaller turndown spool-type variable speed drive. As can be seen in the flow-speed map of Figure 15, the high-speed low-flow corner of the engine requirement is unattainable by the pump-drive combination. Employment of a bypass valve, however, will permit operation in this region.

Because of the small size (0.25 CIPR) of this pump and the low speed at which it runs, the volumetric and mechanical losses, which are not proportional to size, become a greater factor in the overall efficiency, as exhibited in the curves of Figure 16.

3.5.2.2 Variable Displacement Vane Pump

The efficiencies attainable with a variable displacement vane pump in the SES system are shown in Figure 17.

The best efficiencies of a variable displacement vane pump occur at the higher relative displacements. Balancing the pump size against the low capacity requirements of this application necessarily entails slow pump speeds, which further tends to reduce the overall efficiency. A 0.47 CIPR maximum displacement pump was selected with a 1.5 speed increase in the engine gearbox. The best efficiencies obtainable were still generally low.

It does not seem that the SES system warrants selection of such complex equipment.

3.5.2.3 Fixed Displacement Vane Pump with Bypass

The low feed pump capacity required by the SES system justifies exploration of a fixed displacement pump without speed or displacement variation. Figure 18 depicts the power curves for a simple bypassing vane pump of 0.47 CIPR displacement.

Although efficiency suffers from the lost work of bypassing, the actual power loss is quite low (less than 5 horsepower at the worst condition). For this reason and because of the simplicity of this method, the system may well be able to tolerate such a power loss.

3.5.2.4 Two-Stage Vane Pump with Bypass

To reduce the losses in a bypassing system, the pump may be staged in a manner similar to that shown in Figure 10. At the low speed, high flow points, the two elements will be operating in parallel to meet the engine demand. As engine speed increases output from the direct-driven pump far exceeds requirements. At some appropriate signal, a valve can be actuated which will direct the discharge of one element to the inlet of the other; thus, putting both elements in series operation.

Half of the full pressure rise of the system will be carried by each stage. The maximum power loss is thus diminished by 20% in this arrangement, where the displacement of each stage is 0.44 CIPR with the pump running at expander speed. The power curves for this pump are shown in Figure 19.

The switching point will be a function of pump capacity, system considerations and the nature of the control mechanism. However, it is suggested that consideration be given to a simple differential flow sensor that will switch the pump from the parallel to the series mode when bypass flow exceeds engine demand flow.

3.5.2.5 Fixed Displacement Vane Pump with Inducer Bypass

Another unique method of reducing the losses in a bypassing system is the recovery of some of the lost energy in a jet inducer. The bypass function can be performed within the inducer by using a pintle nozzle. This concept is developed on Page E-7 of Appendix E. The power curves for this arrangement are shown in Figure 20 as a function of flow. Figure 21 shows a comparison of the power requirements of this method, with respect to those of the simple bypassing pump discussed above (3.5.2.3). Both pumps are the same size and are running at the same speeds.

3.5.3 Organic Fluid Turbine - ALRC

The high flow rates of the ALRC system place more of a burden on the feed pump to minimize losses.

The gear and vane pump concepts which have been studied are presented below.

3.5.3.1 Fixed Displacement Gear Pump on a Variable Speed Drive

As seen in the performance envelope of Figure 22, a 0.9 cubic inch per revolution (CIPR) gear pump on a ball-type variable speed drive, having a 9:1 speed ratio range, is, by itself, unable to fulfill the complete engine flow envelope because of the extremely high turndown required. An accumulator can accommodate the transients which occur in the low-speed high-flow corner. The efficiencies accruing to this configuration fail to meet specification limits as indicated in Figure 23.

3.5.3.2 Fixed Displacement Vane Pump on a Variable Speed Drive

A vane pump of the same size as above on the same variable speed drive exhibits better efficiencies over the full flow range (see Figure 25) because of its inherent flatter efficiency characteristic. (See Appendices C and D.) In

Figure 24, it is noted that the gearbox ratio must be adjusted slightly to bring in some of the low-flow points which had fallen out of the range of the variable speed drive because of the better vane pump volumetric efficiency at low speed.

3.5.3.3 Two-Stage Gear Pump on a Variable Speed Drive

To better utilize the full range efficiency capability of the gear pump, a two-stage series-parallel arrangement was explored. The arrangement will be similar to that shown in Figure 10 and its function will be essentially as described in Section 3.5.1.3. The displacement of each element will be 0.6 CIPR. This configuration also permitted the use of the lower turndown, higher efficiency, spool-type variable speed drive. However, as indicated in Figure 26, the extreme corners of the flow envelope are still inaccessible. This can be solved by using an accumulator at high flow and a dump valve at low flow.

A substantial improvement in efficiency is noted from Figures 27 and 28. The switching point is important in maintaining high efficiencies at all conditions.

3.5.3.4 Two-Stage Vane Pump on a Variable Speed Drive

A further improvement in full range efficiency can be obtained by using a vane pump in a series-parallel arrangement. This pump also comprises two 0.6 CIPR elements. As observed in Figure 29, the unattainable low-speed high-flow corner of the flow envelope is so minimal that it probably can be neglected. A dump valve can be used at the low flow points. Efficiencies are shown in Figures 30 and 31.

3.5.3.5 Variable Displacement Vane Pump

The efficiencies accruing to a variable displacement vane pump in the ALRC system are shown in Figures 32 and 33. The maximum displacement of this pump is 1.1 CIPR.

3.6 Materials

The materials utilized in the construction of the various vapor generator feed pumps have been selected on their ability to perform adequately in the specific working fluids. The selections were based on material compatibility data received from the producers and users of Fluorinol 85 and AEF-78, as well as on tests run by Chandler Evans.

Main housings will be constructed from cast 356 aluminum or gray cast iron. For the water feed pump, only cast 356 aluminum will be considered. Anodizing will be used to provide additional corrosion resistance for the cast aluminum.

Impellers will be constructed from cast 356 aluminum and, as in the case of the aluminum housings, will be anodized to afford additional corrosion protection.

Pumping gears, with their associated drive trains, will be manufactured from either nitrided AMS 6470 (used extensively in Chandler Evans fuel pumps) or from carburized SAE 8620. The drive splines will be chromium plated to provide additional wear resistance. In the case of the vane pump working on water, the drive shaft will be fabricated from Greek Ascoloy, a corrosion resistant steel.

AISI 420 stainless steel, hardened to Rockwell C 48-53, and AISI 52100 steel, hardened to Rockwell C 60, have been selected as candidate materials for vane pump rotors. The former alloy will provide the necessary corrosion resistance for operation in the water environment. An air hardening tool steel has also been considered as a rotor material for the organic fluid feed pump.

Carbon-graphite, tool steel and Stellite are considered prime candidates for vanes. In the case of the carbon graphite, a special grade has been selected with a coefficient of expansion that matches that of the rotor to minimize fluid losses at elevated temperatures. This problem will not exist in the case of the tool steel and Stellite vanes. Tool steel is not recommended for the water system because of its lack of corrosion resistance. Other materials that might be considered for vanes are S-monel and glass-filled polyimide (for the water system).

440C Stainless steel and 52100 steel have been selected as liner materials for the vane pumps. The 440C will be hardened to Rockwell C 53-60, and the 52100 steel to Rockwell C 60. 52100 has been eliminated as a liner material for the water feed pump because of its lack of corrosion resistance.

Rotational wear tests have been conducted on a carbon graphite/440C stainless steel system to ascertain the suitability of this combination for use as vane and liner, respectively. The results are detailed under Bearing Material Tests, Page 55.

In the gear and vane pumps presented in this report, the pumping element bearings are lubricated by the working fluid. Only resin impregnated carbon graphite is recommended for these bearings in both the Fluorinol 85 and water. This material may also be used in the AEF-78 in addition to leaded bronze. As demonstrated by Chandler Evans testing, each set of materials and fluids has a critical load-speed limit. Bearing designs will be based on this information.

The selection of suitable seal materials has required careful consideration and because of the variety of fluids involved, the materials utilized for one fluid may be unsatisfactory for another. Ethylene propylene rubber is recommended for use in Fluorinol 85 and FC75, Viton (fluorocarbon) for use in perchloroethylene, neoprene for use in AEF-78 and nitrile for use in water.

Tables III and IV summarize the material selection for vane pumps and gear pumps, respectively.

Appendix G provides data on the physical properties of the loop fluids in comparison with the properties of CECO used fuels and other similar fluids.

3.7 Test Programs

3.7.1 Micro-Ryder Testing of Nitrided and Carburized Steel Gears

Pumping gear evaluations for gear loading capacity and gear life have been conducted with water. Fluorinol 85 and AEF-78, and compared with JP-4 using AMS 6260 gears.

The tests were carried out with the Micro-Ryder gear machine which is run at a constant speed and gear geometry while varying the gear loading torque in incremental steps.

Base line running was conducted on nitrided AMS 6470 gears in MIL-J-5624. JP-4 aviation fuel. since voluminous service history information is available on this combination.

A standard gear material, carburized AMS 6260, was utilized to evaluate the relative scoring and wear resistance of the various system fluids.

The relative scoring is shown in Figure 34 which shows percent of tooth area scuffed versus machine load. A lower amount of scuffing indicates that a higher gear tooth load is possible for a given life requirement.

The maximum scoring factors (Pressure x Velocity x Length, PVT) for gear pump design, using typical pump gear materials, are known. Figure 34 indicates that both FL-85 and AEF-78 are good fluids for gear pump applications. On the other hand, we see that water is not as readily adaptable.

Using limiting PVT factors for the various gear materials selected, a series of gear pump parameters can be established, biased by the scuff resistance of the fluid used, as determined by such testing reported herein.

3.7.2 Elastomer Testing at TECO in Fluorinol 85

Samples of both fluorosilicone and silicon rubber seals were sent to TECO for their test and evaluation in Fluorinol 85. Their test report is included as Appendix B of this report.

3.7.3 Elastomer Testing at ALRC in AEF-78

Samples of both fluorosilicone and silicone compounds were sent to ALRC for their test and evaluation in AEF-78.

These seal materials were placed in a container of AEF-78 at ambient temperature. Air was evacuated from the container preparatory to raising the temperature. However, the seals began to deteriorate immediately as indicated by discoloration of the fluid. The test was halted, since it was obvious that AEF-78 is a solvent for fluorosilicone and silicone.

3.7.4 Pump Testing

3.7.4.1 Gear Pump

A Chandler Evans Model MFP-150 Main Fuel Pump was tested on MIL-T-5624 JP-4, isooctane, Fluorinol 85 and and perchloroethylene (a recommended substitute for AEF-78). Since the MFP-150 has a history of pumping JP-4 and isooctane, the testing with these aviation fuels was used as a basis of comparison for the other fluids.

Referring to the fluid property data contained in Appendix G, wherein the process fluid viscosities were equal to or greater than the aviation fuel viscosities, it was expected that the volumetric performance with the process fluids would be at least as good as the performance with the aviation fuels.

The volumetric performance noted during the pump's operation on Fluorinol 85 at low speeds and pressures at the beginning of the test cycle was as good as or better than the prior calibration on MIL-J-5624, Grade JP-4. The pump performance did not deteriorate significantly until later in the cycle during the highest discharge pressures and pump speeds. Subsequent running with perchloroethylene showed a pronounced decrease in the overall efficiency due mainly to a loss of mechanical efficiency. See Figure 35.

The detail parts inspection after test revealed severe pumping gear bearing face wear with moderate journal bearing wear. These observations tend to support the pump's performance loss due to mechanical efficiency degradation.

The gear tooth flanks remained in good condition while running on the Fluorinol 85, which further confirmed the Micro-Ryder gear test results reported on Page 44.

The gear tooth flanks were badly scuffed after running on perchloroethylene, which points to the poor lubricity characteristics of this fluid. Since lubricity is a function of chemical composition, and perchloroethylene differs chemically from AEF-78, an accurate assessment of a gear pump in AEF-78 will require a test in that fluid.

The test pump was fitted with new pumping gears and bearings and was returned to test to evaluate the NPSH (net positive suction head) characteristics of the impeller boost stage in Fluorinol 85. The impeller performance is shown on Figure 36. The test data indicates that the state-of-the-art designs will adequately fulfill the requirements of each of the Rankine Cycle Systems.

As far as the Fluorinol 85 is concerned, the problem in defining a gear pump to handle this fluid resolves to the development of suitable bearings.

This appraisal also applies to the perchloroethylene, plus the need for selection of a gear material and geometry to overcome the tooth scuffing.

The Chandler Evans MFP-150 pump which has been used as a "base line" for evaluating gear pumps in both the TECO and ALRC systems was designed for operation over a pressure range of 200 to 860 psi. Since, in these Rankine cycle applications, the pressures are maintained at consistently higher levels for all operating conditions, the area under pressure loading can be reduced substantially while still assuring adequate gear face sealing.

Thus, it is estimated that the gear thrust loading which appears to be the major obstacle in running the MFP-150 pump in Fluorinol 85 and perchloroethylene can probably be decreased by a factor in the order of 2 to 1. Another approach to alleviating thrust overload would be employment of a fixed-clearance gear package.

In the matter of radial loads, which do not seem to be as severe as the thrust loads, the journal can be enlarged in accordance with the criterion stated on Page C-6 of Appendix C. The diameter of the journal bearing can safely be increased by 9% with a consequent radial load reduction of 8%.

The use of helical pumping gears will further reduce radial loading. as well as providing more tooth contact which will tend to prevent the scuffing observed from pumping perchloroethylene.

The Model MFP-150 pump used in the testing described above was fitted out with carbon-graphite pumping gear bearings for a preliminary evaluation of performance on Fluorinol 85 and AEF-78.

The efficiencies shown in Figure 37 are comparable to those obtained from operating this pump on JP-4 aviation fuel. It should be pointed out, however, that in the several tests and rebuilds to which this pump had been subjected, the efficiencies would be expected to suffer. The data plotted in Figure 37 is in the normally low-efficiency range of this particular pump. Of interest is the improvement of overall efficiency with running on AEF-78.

The use of carbon graphite will dictate certain design practices, such as no sharp corners and no thin sections. However, to expedite this testing in an existing pump, the carbon bearings were fabricated to the same configuration as the bronze bearings for which the pump was originally designed. During the testing one of the pressure loaded bearings fractured at a sharp corner. In spite of this failure, no significant wear was exhibited on either the thrust or journal bearing surfaces.

The results of this testing encourages us to feel that carbon bearings can be readily adapted to this application.

3.7.4.2 Variable Displacement Vane Pump

A variable displacement vane pump was delivered to Aerojet Liquid Rocket Company to be run in their test loop on AEF-78. ALRC estimates that this testing will commence on or about December 15, 1972.

These tests, in addition to determining the adaptability of this pump to the Aerojet system, will provide a comparison of performance in AEF-78 with respect to aviation fuel.

3.7.3 Bearing Material Tests

A series of bearing wear tests were conducted on several candidate bearing materials in each of the working fluids. The results of these tests are intended to set load-speed envelopes for both bearing and vane design purposes.

The tests were performed on Chandler Evans' wear test rig in which a rotating upper specimen is loaded against a fixed lower specimen, with both specimens immersed in a bath of the working fluid. Both the load and the speed are variable. The tests. which lasted for periods of 30-45 minutes, were run at room temperature, but during the test, the fluid temperature rose, in some cases exceeding 120°F. The results of these tests are summarized below.

Fluorinol 85

(a) Leaded Bronze vs. Nitrided Nitralloy

Lower Specimen - 20% leaded bronze and Electrofilm solid film lubricant

Upper Specimen - Nitrided Nitralloy (AMS 6470)
Case Depth - .012" Min.
Hardness - Rockwell 15N. 92 Min.

Duration of Test - 45 Minutes

Testing on this combination was restricted to one test at 135 PSI load and 1550 fpm. producing a PV of 209,250. Examination of the wear surfaces showed no signs of distress. Further testing of this system was abandoned when it was discovered that Fluorinol 85 had an adverse reaction with the leaded bronze (see Page 61).

(b) Resin Impregnated Carbon Graphite vs. Nitrided Nitralloy

Lower Specimen - Graphitar 39 (U. S. Graphite)

Upper Specimen - Nitrided Nitralloy (AMS 6470)
Case Depth - .012" Min.
Hardness - Rockwell 15N, 92 Min.

Duration of Test - 30 Minutes

The test results indicate that the PV should not exceed 930,000. A load-speed envelope is shown in Figure 38.

(c) Resin Impregnated Carbon-Graphite vs. 440C Stainless Steel

Lower Specimen - 440C Stainless Steel
Hardness R_C 55

Upper Specimen - Graphitar 113 (U. S. Graphite)

Duration of Test - 45 Minutes

The test results indicate a maximum allowable PV of 570,240. A load-speed envelope is shown in Figure 39.

AEF-78

(a) Leaded Bronze vs. Nitrided Nitralloy

Lower Specimen - 20% Leaded Bronze and Electrofilm solid film lubricant

Upper Specimen - Nitrided Nitralloy (AMS 6470)
Case Depth - .012" Min.
Hardness - Rockwell 15N. 92 Min.

Duration of Test - 30 Minutes

The test results indicate an allowable PV range in excess of 2.378.000. Typical sleeve bearing PV values encountered in Chandler Evans pumps are usually well below this value. A load-speed envelope is shown in Figure 40.

(b) Resin Impregnated Carbon Graphite vs. 440C Stainless Steel

Lower Specimen - 440C Stainless Steel
Hardness R_C 55

Upper Specimen - Graphitar 113 (U. S. Graphite)

Duration of Test - 30 Minutes

The test results indicate that the PV should not exceed 570.240. A load-speed envelope is shown in Figure 41.

Water (Distilled)

(a) Resin Impregnated Carbon Graphite vs. 440C Stainless Steel

Lower Specimen - 440C Stainless
Hardness - R_C 55

Upper Specimen - Graphitar 113 (U. S. Graphite)

Duration of Test - 30 Minutes

The test results indicate an allowable PV range up to at least 570,240. A load-speed envelope is shown in Figure 42.

Perchloroethylene

(a) Leaded Bronze vs. Nitrided Nitralloy

Lower Specimen - 20% Leaded Bronze and Electrofilm solid film lubricant

Upper Specimen - Nitrided Nitralloy (AMS 6470)
Case Depth - .012" Min.
Hardness - Rockwell 15N. 92 Min.

Duration of Test - 45 Minutes

The test results indicate an allowable PV range up to at least 634,725. A load-speed envelope in shown in Figure 43.

(b) Resin Impregnated Carbon Graphite vs. 440C Stainless Steel

Lower Specimen - 440C Stainless Steel
Hardness R_C 55

Upper Specimen - Graphitar 113 (U. S. Graphite)

Duration of Test - 45 Minutes

Only one test was performed on this combination at a load of 510 PSI and a speed of 2400 fpm, producing a PV of 1,224,000. Discoloration of the 440C track showed that this condition was marginal. No further tests were run on this material system.

In commenting on allowable PV values for vane tips, it is Chandler Evans experience that calculated PV values using vane tip Hertz loads can exceed those predicted by wear test results by a substantial margin, without any noticeable evidence of wear.

Testing of the MFP-150 pump, which showed excessive bearing wear while running on Fluorinol 85, also produced a discoloration of the test fluid and prompted an investigation into the effect of Fluorinol 85 on leaded bronze. Immersion tests were conducted on samples of leaded bronze with and without the dry film lubricant in Fluorinol 85 at room temperature. After 2-3 hours. the fluid became milky and with continued exposure. a heavy white precipitate was observed. Analysis of this precipitate showed that it was rich in lead and it was concluded that Fluorinol 85 was not compatible with alloys containing lead. As a result of these findings. Halocarbon. the producers of Fluorinol 85. was informed and additional work by Halocarbon confirmed Chandler Evans test results.

Immersion tests of leaded bronze in AEF-78 at room temperature showed no reaction after 72 hours.

Immersion tests of resin impregnated carbon graphite in boiling Fluorinol 85 also showed no reaction after 48 hours.

3.8 Conclusions

3.8.1 Variable Speed Drives

The variable displacement traction drives included in the conceptual models developed in this study are rugged, simple and reasonably efficient. They deserve consideration in a variable delivery pumping system.

There is activity currently taking place in this field toward increasing speed range and efficiency. Such advances will further enhance the position of variable speed drives in this application.

3.8.2 Pumps

3.8.2.1 General

None of the pumps analyzed in this study, be they the conceptual models developed by Chandler Evans or units under development by the system contractors, meets the full range of efficiencies specified by this contract (50% at 10% flow and 70% between 30% and 80% flow). The arrangement that will most nearly fulfill specification limits is a two-stage vane pump on a variable speed drive.

3.8.2.2 Gear Pumps

Gear pumps will perform very reliably in both Fluorinol 85 and AEF-78. Carbon graphite bearings will function satisfactorily in both fluids, while leaded bronze will also serve the AEF-78.

A gear pump is not recommended for water.

There are development activities underway at Chandler Evans to improve gear pump efficiencies. such as the use of helical gears and the employment of fixed clearances. Success in these efforts may prove gear pumps to be viable candidates for the organic fluid Rankine cycle feed pumps.

However, the current generation of gear pumps will not meet the efficiency requirements specified in this contract.

If gear pump efficiencies can be raised to acceptable levels. they would be an ideal pump for this application because of their inherent ruggedness and low cost.

3.8.2.3 Vane Pumps

The vane pumps investigated in this study will be run at sufficiently low speeds to permit the use of low density vane materials; thus reducing operating losses. Although not considered in the fixed displacement vane pumps presented in this report. a further improvement in efficiency may be obtained by using hydrodynamic vane tips.

The basic efficiency characteristics of vane pumps recommend them for prime consideration as Rankine cycle feed pumps.

3.8.2.4 Variable Displacement Pumps

The nature of the performance envelopes for these Rankine cycle feed pumps is such that a variable displacement pump must function in the low percentage range of full displacement at most of the operating conditions. Efficiencies are low under these circumstances. For this reason and because of complexity and cost, variable displacement vane pumps are not recommended.

The mechanical problems of sealing the working fluid from the lubricant coupled with the size of a piston pump have led us to view this type of pump unfavorably for handling the low viscosity Rankine cycle working fluids.

3.8.3 Specific Pumps and Drives

3.8.3.1 Organic Fluid Reciprocator - TECO

The best efficiencies are obtained by a two-stage vane pump on a

6.25:1 turndown variable speed drive. The low-flow efficiencies are especially good.

However, a simple vane pump on a variable speed drive exhibits reasonably good efficiencies and may warrant consideration on the basis of simplicity.

3.8.3.2 Water Reciprocator - SES

The low capacity of the feed pump in the water system with its concomitant low power demands does not seem to justify investment in a variable delivery pump.

Therefore, a fixed displacement bypassing pump is recommended. The simplest and most efficient over the full operating range is a vane pump bypassing through a jet inducer with a pintle nozzle serving as the bypass valve.

We have limited our attention to vane pumps because of material restrictions.

3.8.3.3 Organic Fluid Turbine - ALRC

A two-stage vane pump on a 6.25:1 turndown variable speed drive comes close to meeting the specified efficiency requirements at all operating conditions.

3.8.4 Inducers

Inducers are needed by all of the pumps presented herein. They will be capable of handling the expected net positive suction heads (NPSH) at pump inlet.

4. INSTALLATION STUDIES

Installation, size and envelope data have been prepared for the various power plants under study based on the available information at the time of preparation. It was noted during our contacts with the system contractors that these power plants are, in general, going through continual installation and envelope changes. Although the updated versions of our drawings will be to latest known requirements of space and mounting, it is noted that any future effort, after definition of concept, must be more closely coordinated with and made a part of any engine program where the results of this study are applied.

We have tabulated below by Company, Drawing Number and Description, the installation study effort to date. Copies of these drawings have been transmitted to the engine contractors and to the EPA.

The salient pump features have been extracted from several of these layouts and are included herein as Figures 44 through 50. These illustrations show the basic design concepts and give approximate envelope dimensions. Some modification of component arrangement and space envelope is possible in these pumps to accommodate engine requirements.

4.1 Thermo Electron Corporation

L1640J2	Gear pump on sheave type variable speed drive with differential - revised engine drive housing.
L1640J3	Gear Pump on sheave type variable speed drive with differential - adapting to existing engine drive housing.
L1640J5	Gear Pump on Sundstrand hydrostatic variable speed drive.
L1640J6	Gear pump on TPI traction type variable speed drive.

L1640J7	Gear pump on TPI traction type variable speed drive with differential.
L1640J8	Gear pump on TPI traction type variable speed drive with differential sized for added power requirement of differential.
L1640J9	Gear pump on Cleveland Worm & Gear ball-type variable speed drive.
L1640D10) L1640D11) L1640D12)	Variations of a two-stage fixed displacement gear pump on a variable speed drive.

4.2 Steam Engine Systems

L1643A3	Cross section of 2.5 GPM vane pump.
L1643J5	Vane pump on TPI traction type variable speed drive with differential.
L1643D6	Vane pump on TPI traction type variable speed drive (no differential).

L1643D7	Vane pump sized to meet SES limiting condition and bypass at other conditions.
L1643D8	Two-stage fixed displacement vane pump.

4.3 Aerojet Liquid Rocket Company

L1642D1	Variable displacement vane pump designed for ALRC envelope.
L1642D2	Variable displacement vane pump in bar stock housing.
L1642J3	Layout showing installation features of a variable displacement vane pump prepared for delivery to ALRC.
L1642J4	Gear pump on TPI traction type variable speed drive.
L1642D5	Vane pump on TPI traction type variable speed drive.

L1642D6	Two-stage fixed displacement vane pump.
L1640J9	Gear pump on Cleveland Worm & Gear ball-type variable speed drive.
99139X	Assembly drawing of variable displacement vane pump supplied to ALRC for testing.

TABLE I - VARIABLE SPEED DRIVES

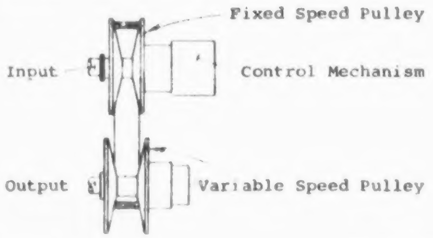
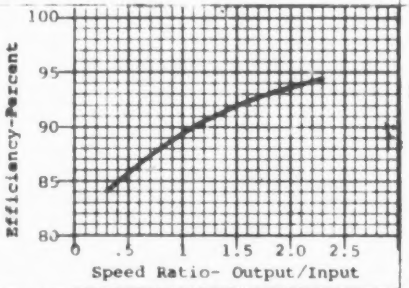
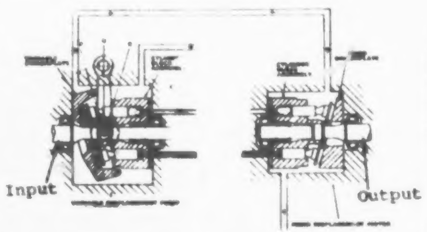
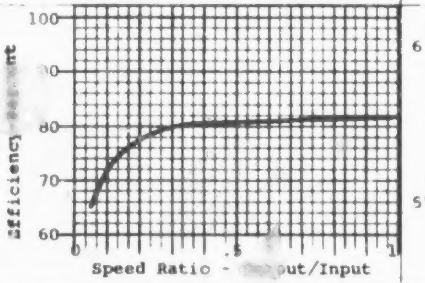
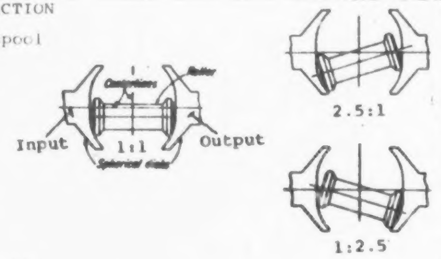
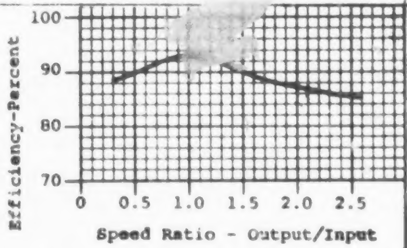
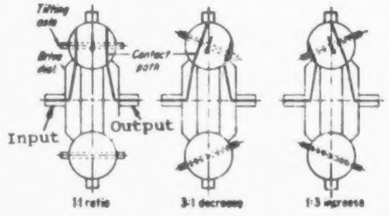
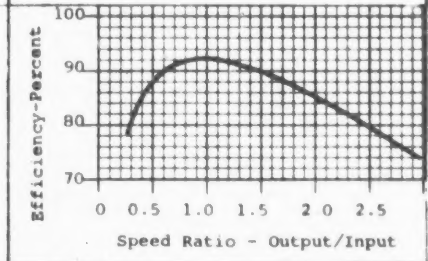
DRIVE TYPE	SPEED RATIO RANGE	EFFICIENCY	SIZE 12 H.P.	CONTROL	STATE OF DEV.	MANUFACTURERS CONTACTED
VARIABLE PITCH SHEAVE  Fixed Speed Pulley Input Control Mechanism Output Variable Speed Pulley	1:36 to 2.2:1 (8:1)	 	10"x 23" x9"	Dis-placement	Available	Llewellen Mfg. Co. Speed Selector, Inc. Winsmith Division UMC Industries Gerbing Mfg. Co. Reliance Electric Co.
HYDROSTATIC  	0:1 (Inf.)	 	PUMP 6"x 6" x7" MOTOR 5" x 7" x 6"	Dis-placement	Available	Sunstrand Hydro-Transmission Division Hydeco Division General Signal Corp.
TRACTION Spool  Roller Input 1:1 Spherical disks Output 2.5:1 1:2.5	1:2.5 to 2.5:1 (6.25:1)	 	6" DIA. x 10"	Force	Prototype	Traction Propulsion Inc.
Ball  Tilting axle Drive disk Contact path Input Output 1:1 ratio 3:1 decrease 1:3 increase	1:3 to 3:1 (9:1)	 	10" DIA. x 7"	Dis-placement	Available Design to be Modified	Cleveland Worm & Gear Division Eaton Corp.

TABLE II

	RECIP.-ORGANIC (TECO)	TURBINE-ORGANIC (ALRC)	RECIP.-WATER (SES)
INSTALLATION			
Fluid	FL-85	AEF-78	Water
Expander Speed Range	300/1800 RPM	16,800/31,200 RPM	400/2500 RPM
Flow Control	Vaporizer Pressure Signal	Accel. Pedal-Metering Orifice	Vaporizer Pressure Signal
Pump and Elevation - 0 Grade (1)	+3 In.	Unknown	+5 In.
Pump and Elevation - 30% Grade (1)	-10/+16 In.	Unknown	-3 1/2/+13 1/2 In.
Minimum Liquid Head at Inlet	10 In.	10 In.	Low
Mounting	Expander Pad	Gearbox Pad	Bracket
Coupling	Adapt to Spline	Spline	V-Belt
Rotation (Pump viewed from Shaft End)	CCW	CW	CCW
Expander to Pump Drive Ratio	1 to 1	1.85 to 1	1 to 1
Inlet Line Size and Length	1 In. ID x 5 Ft.	1 1/4 In. OD x (Unk.)	3/4 In. ID x (Unk.)

(1) Vertical distance from condenser outlet port centerline to pump drive centerline - positive value when pump drive is above condenser port

TABLE II (Continued)

	RECIP.-ORGANIC (TECO)	TURBINE-ORGANIC (ALRC)	RECIP.-WATER (SES)
CAPACITY			
Flow Rate - Min. Speed (N)(2)	7.4 GPM at N = 300 RPM 13.75 GPM at N = 600 RPM	20 GPM at N = 16,800 RPM	0.8 GPM at N = 400 RPM
Flow Rate - Intermediate to Maximum Speed (N)	17 GPM at N = 800 RPM 15.4 GPM at N = 1800 RPM	28 GPM at N ≥ 16,800 RPM	2.5 GPM at N ≥ 1000 RPM
Minimum Flow Rate - All Speeds	0.4 GPM at N = 300 RPM 6 GPM at N = 1800 RPM	1.8 GPM at N = 16,800 RPM 2.0 GPM at N = 31,200 RPM	1 PPM at N = 400 RPM 3 PPM at N = 2500 RPM
Output Pressure	850 PSIA	600 to 1050 PSIA	1200 PSIA
OPERATING INLET CONDITIONS			
Temperature	100 to 250 °F	130 to 240°F	180 to 270°F
Pressure	5 to 90 PSIA	12.2 to 32.5 PSIA	7 to 50 PSIA
START-UP INLET CONDITIONS			
Temperature	-40 to 275°F	-40 to 275°F	32 to 275°F
Pressure(3)	0.5 PSIA	1.0 PSIA	0.09 PSIA

(2) Flow rate varies proportionally with speed from minimum speed to intermediate speed

(3) Function of system temperature

TA[illegible] I[illegible]

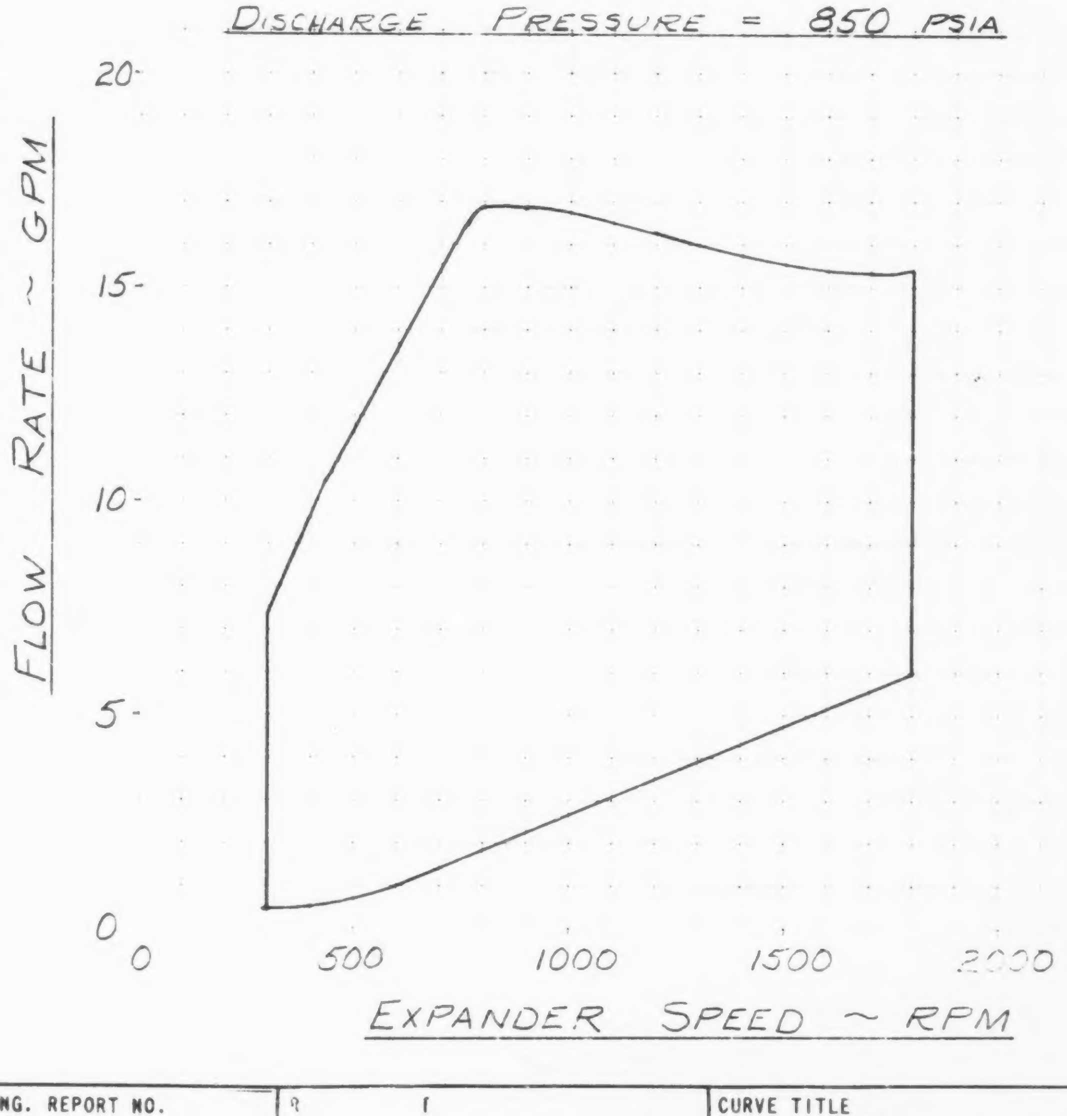
CANDIDATE MATERIALS FOR VAPOR GENERATOR FEED PUMP (VANE PUMP)

Component	WORKING FLUIDS				
	Fluorinol 85	AEF-78	FC-75	Perchloroethylene	Water
Seals	E. P. R. Silicone	Neoprene Butyl Teflon	E. P. R. Nitrile Silicone Butyl Neoprene	Fluorocarbon (Viton) Fluorosilicone	Nitrile Butyl E. P. R. Silicone
Vanes	Carbon-Graphite Tool Steel	Carbon-Graphite Tool Steel	Carbon-Graphite Tool Steel	Carbon-Graphite Tool Steel	Carbon-Graphite Stellite
Bearings	Carbon-Graphite	Leaded Bronze Carbon-Graphite	Leaded Bronze Carbon-Graphite	Leaded Bronze Carbon-Graphite	Carbon-Graphite
Housing (Block)	440C Stainless Steel 52100 Steel	440C Stainless Steel 52100 Steel	440C Stainless Steel 52100 Steel	440C Stainless Steel 52100 Steel	440C Stainless Steel
Side Plates	Carbon-Graphite	Carbon-Graphite Leaded Bronze	Carbon-Graphite Leaded Bronze	Carbon-Graphite Leaded Bronze	Carbon-Graphite
Impeller	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)
Rotor	52100 Steel 420 Stainless Steel Tool Steel	52100 Steel 420 Stainless Steel Tool Steel	52100 Steel 420 Stainless Steel Tool Steel	52100 Steel 420 Stainless Steel Tool Steel	420 Stainless Steel
Drive Shaft	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Greek Ascoloy
Housing, Main	Cast 356 Aluminum Cast Iron	Cast 356 Aluminum Cast Iron	Cast 356 Aluminum Cast Iron	Cast 356 Aluminum Cast Iron	Cast 356 Aluminum (Anodized)

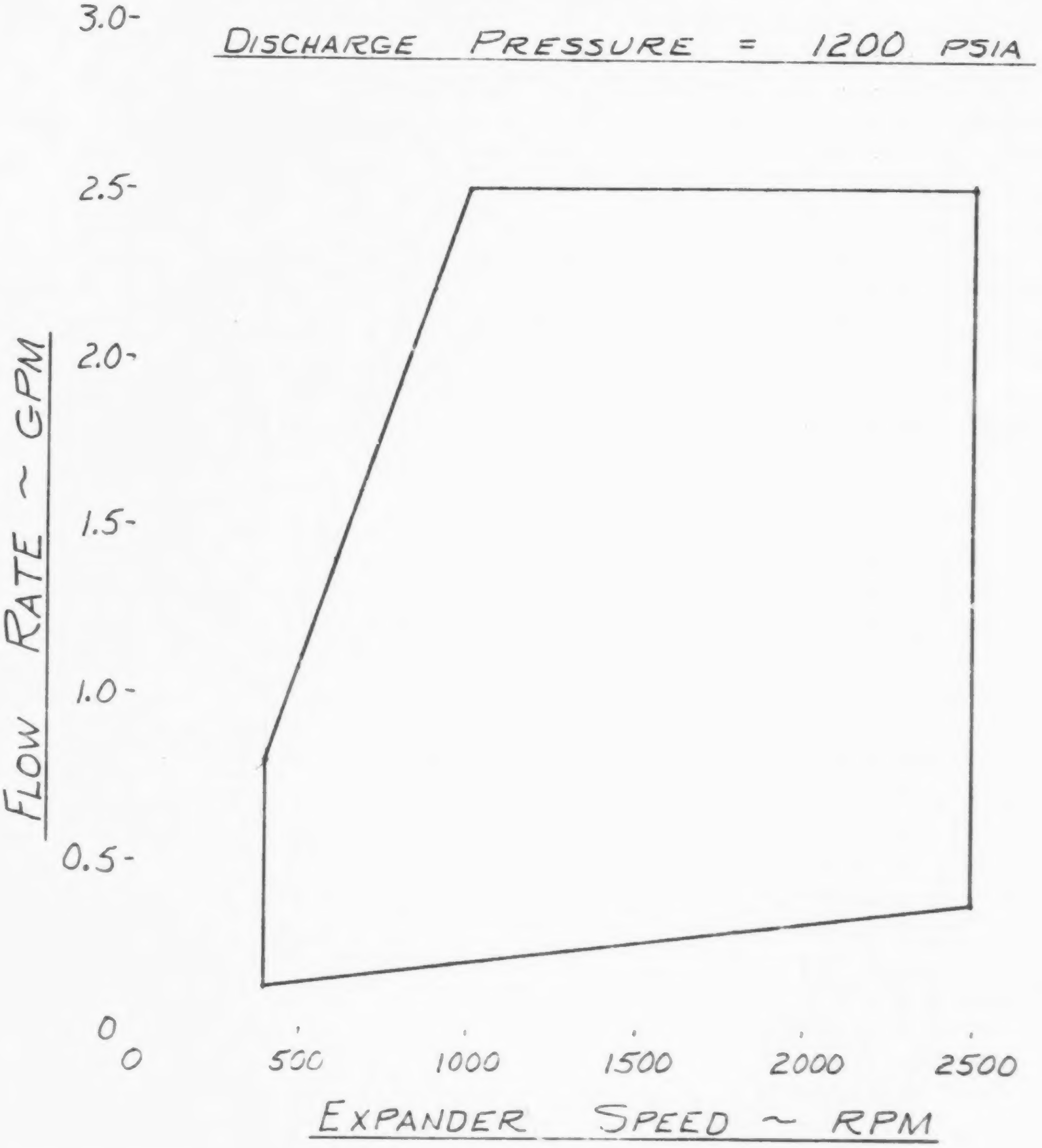
TABLE IV

CANDIDATE MATERIALS FOR VAPOR GENERATOR FEED PUMP (GEAR PUMP)

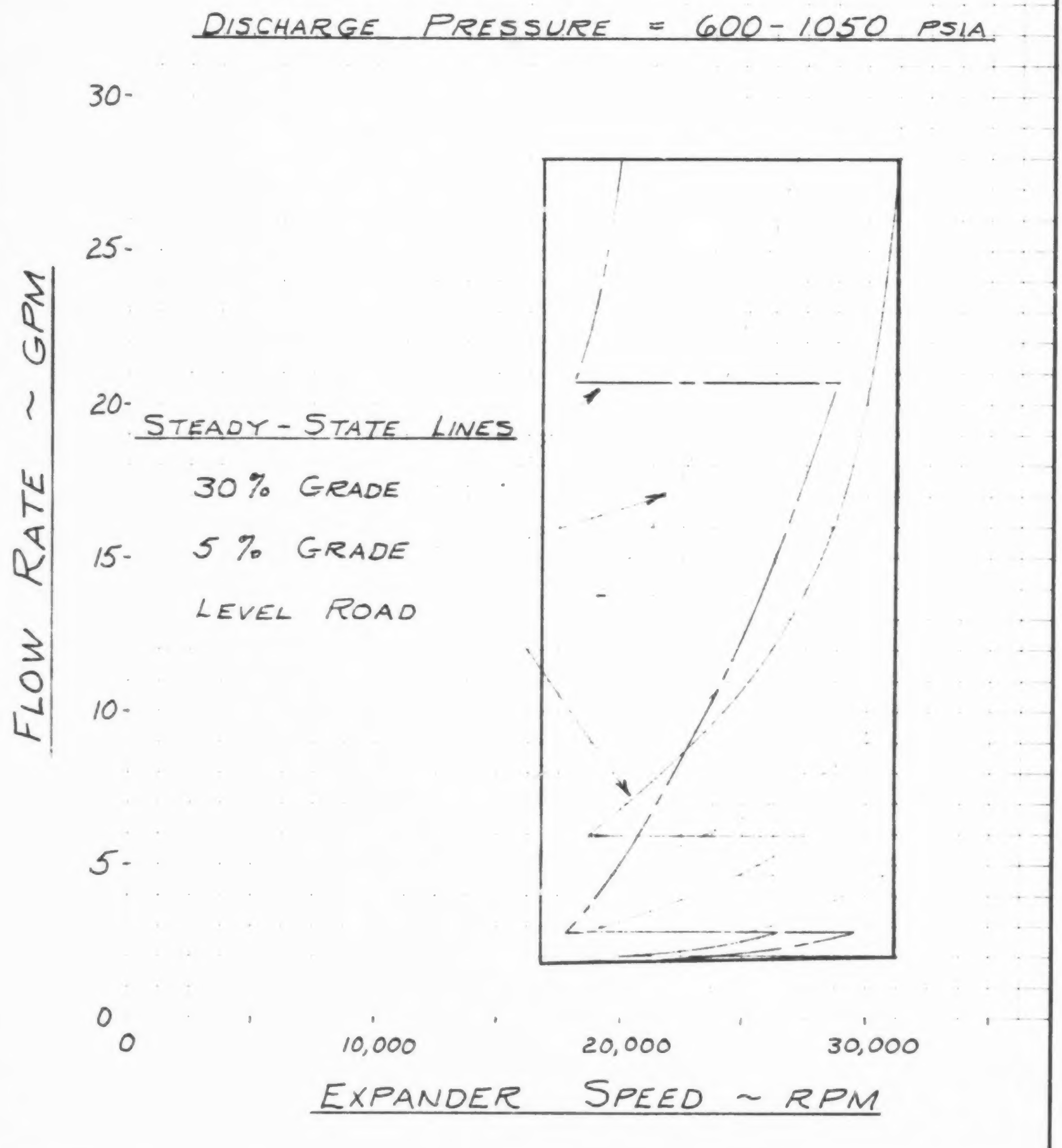
Component	WORKING FLUIDS				
	Fluorinol 85	AEF-78	FC-75	Perchloroethylene	Water
Seals	E. P. R. Silicone	Neoprene Butyl Teflon	E. P. R. Nitrile Silicone Butyl Neoprene	Fluorocarbon (Viton) Fluorosilicone	Nitrile Butyl E. P. R. Silicone
Gears	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	
Bearings	Carbon Graph.	Leaded Bronze Carbon Graph.	Leaded Bronze Carbon Graph.	Leaded Bronze Carbon Graph.	
Housings	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)
Impeller	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)	Cast 356 Aluminum (Anodized)
Drive Shaft	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	Nitrided Nitralloy Carburized SAE 8620	



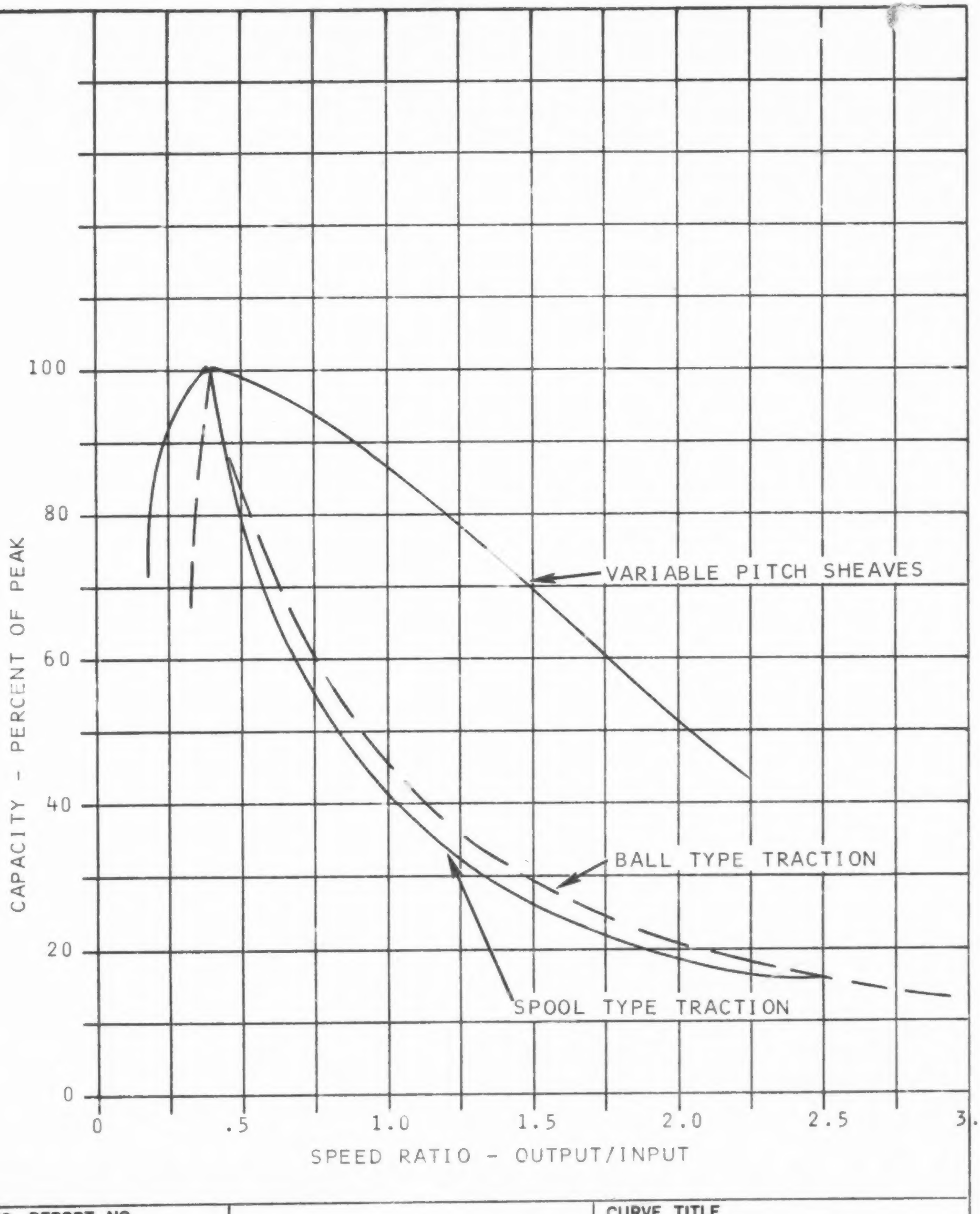
ENG. REPORT NO. APTD-1357	CURVE TITLE FLOW REQUIREMENTS ORGANIC FLUID RECIPROCATOR – TECO
DATE 12-5-72	
PREPARED BY RMR	FIGURE NO.



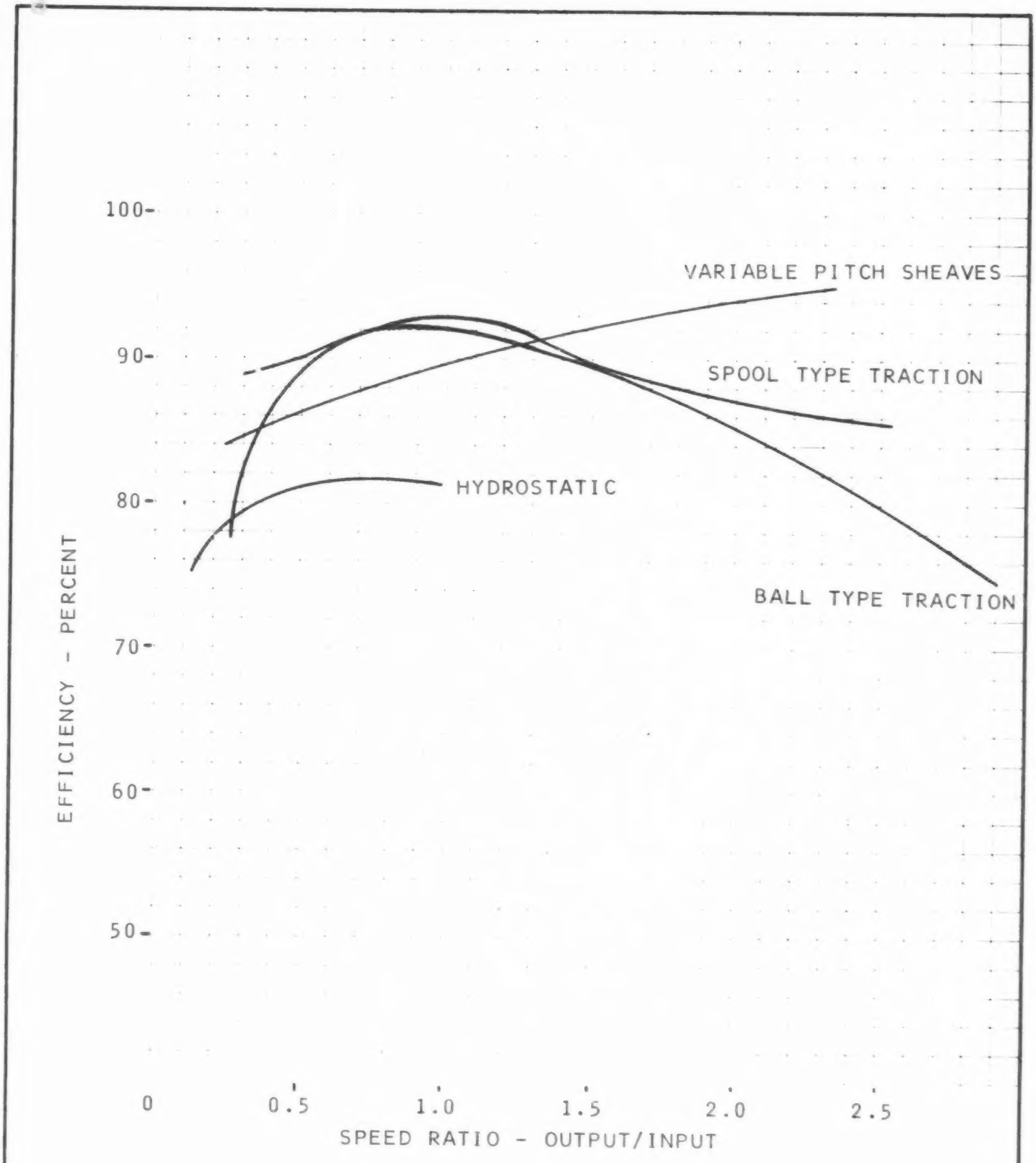
ENG. REPORT NO. APTD-1357		CURVE TITLE FLOW REQUIREMENTS WATER RECIPROCATOR — SES
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. [illegible]



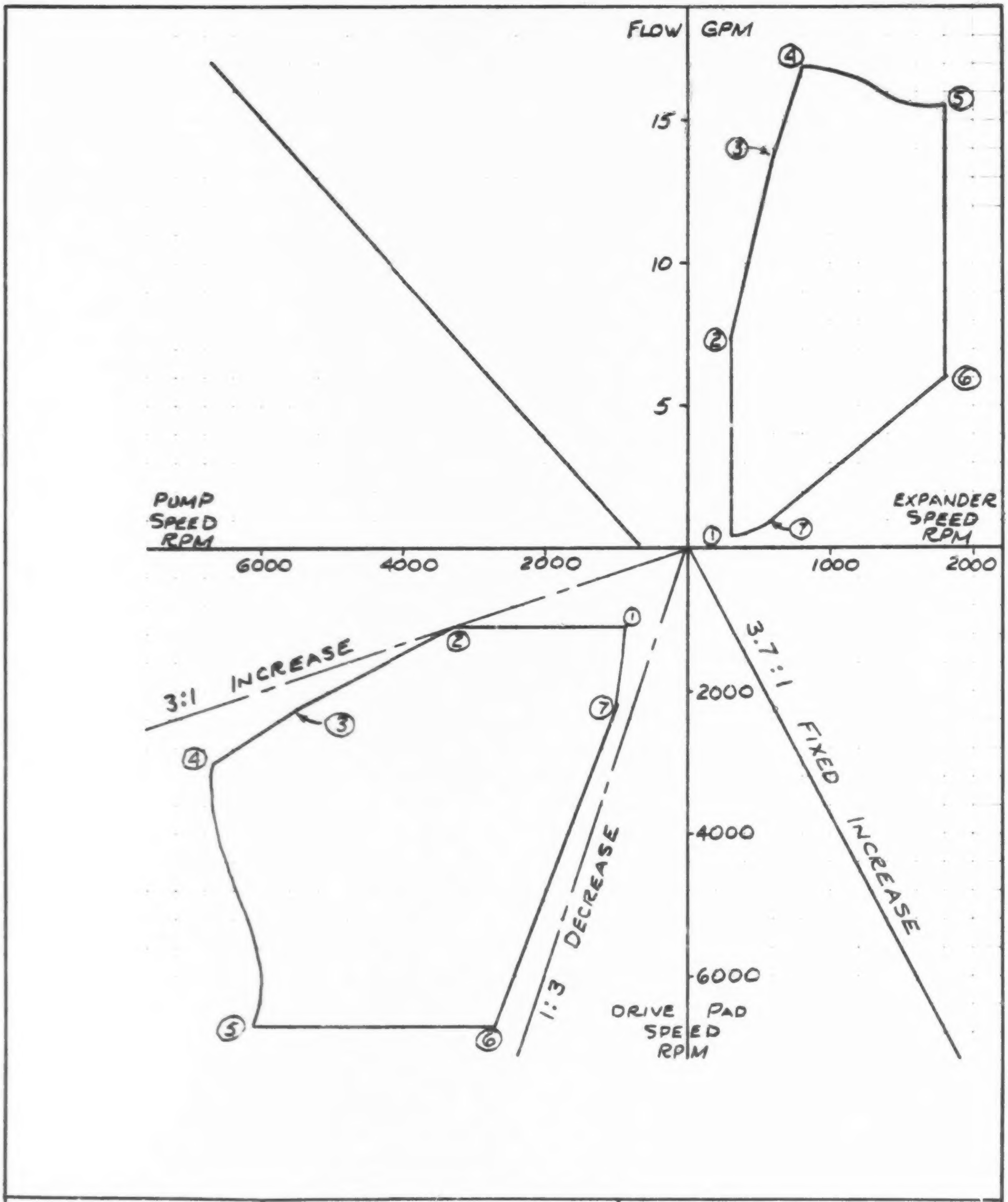
ENG. REPORT NO. APTD-1357		**CURVE TITLE** FLOW REQUIREMENTS, ORGANIC FLUID TURBINE – ALRC
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 3



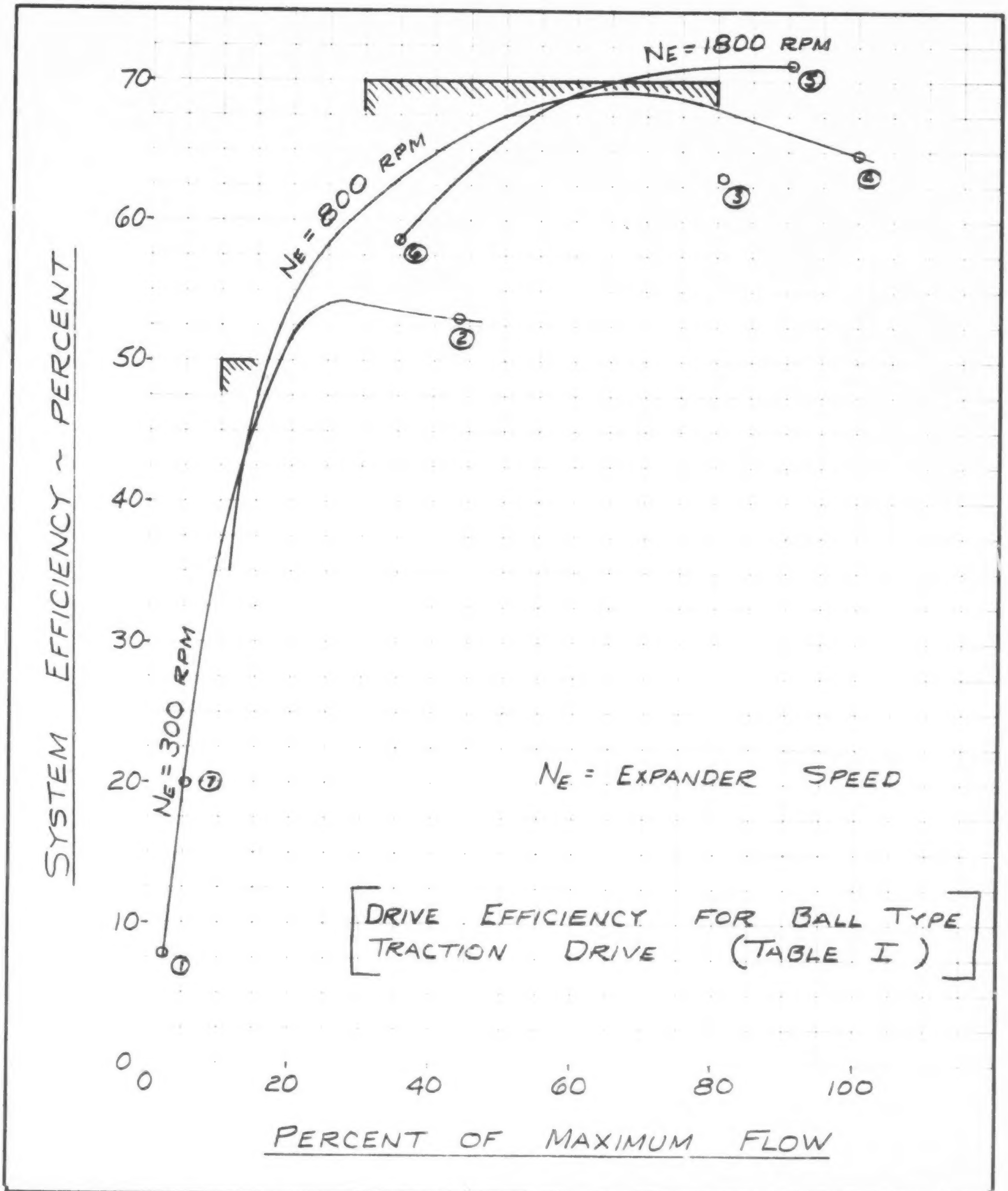
ENG. REPORT NO. APTD-1357	VARIABLE SPEED DRIVES	**CURVE TITLE** TORQUE VS SPEED RATIO
DATE 12-5-72		
PREPARED BY M. R.		**FIGURE NO.** 4



ENG. REPORT NO. APTD-1357	VARIABLE SPEED DRIVES	**CURVE TITLE** EFFICIENCY VS SPEED RATIO
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 5



ENG. REPORT NO. APTD-1357	GEAR PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID RECIPROCATOR (TECO)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 6



ENG. REPORT NO. APTD-1357

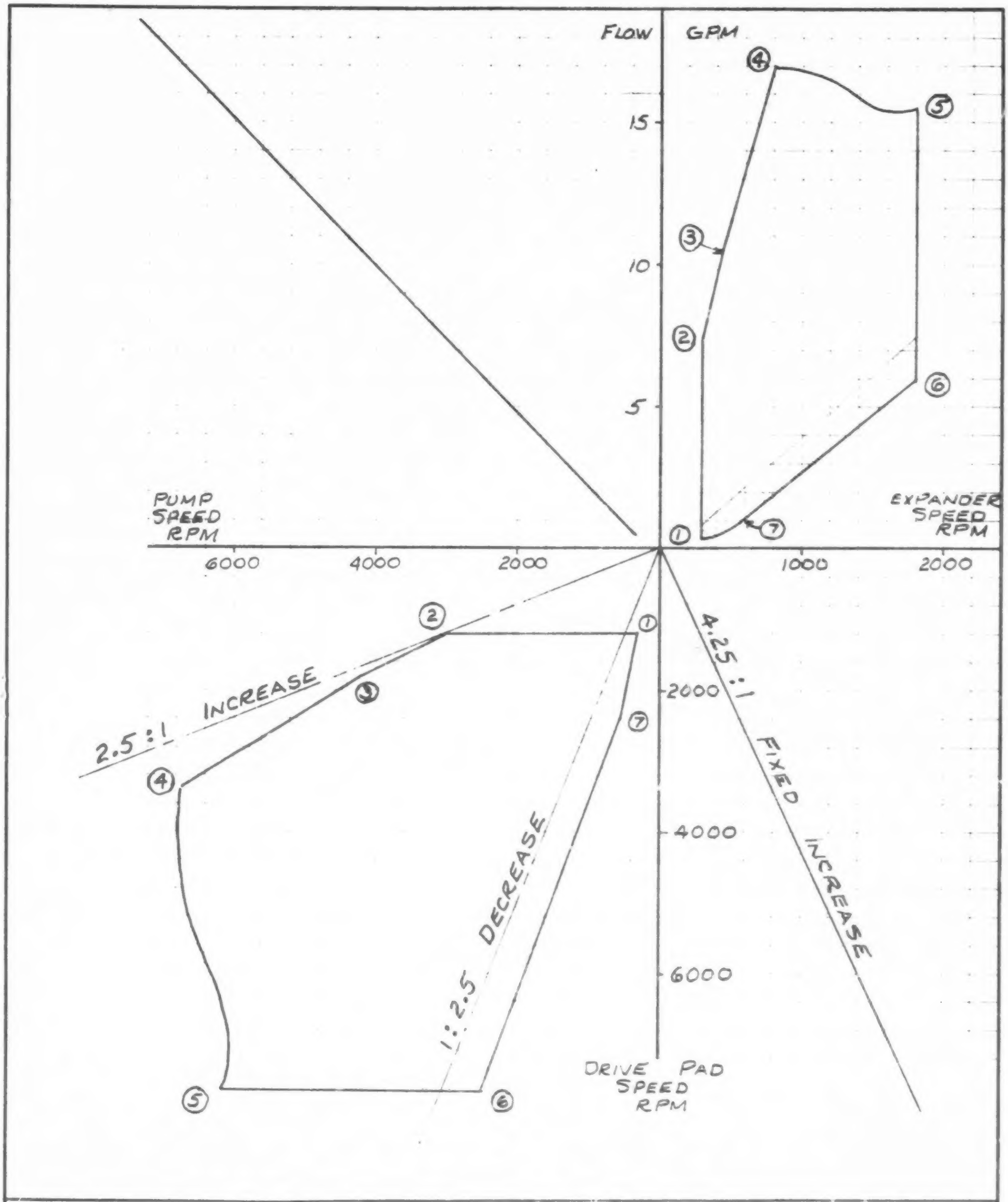
DATE 12-5-72

PREPARED BY RMR

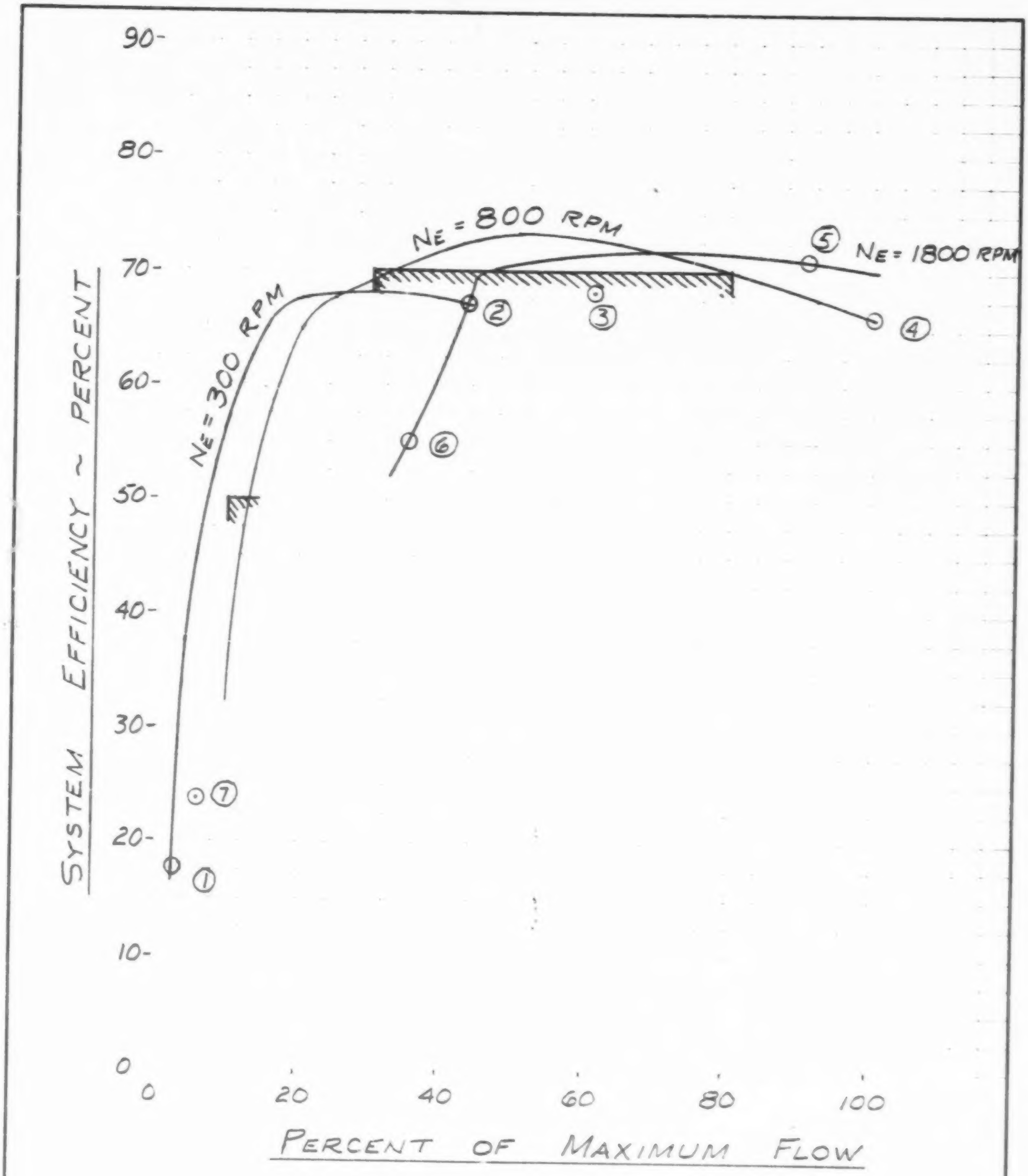
GEAR PUMP ON
VARIABLE SPEED DRIVE
ORGANIC FLUID
RECIPROCATOR (TECO)

CURVE TITLE SYSTEM EFFICIENCY

FIGURE NO. 7



ENG. REPORT NO. APTD-1357	VANE PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID RECIPROCATOR (TECO)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 8



ENG. REPORT NO. APTD-1357

DATE 12-5-72

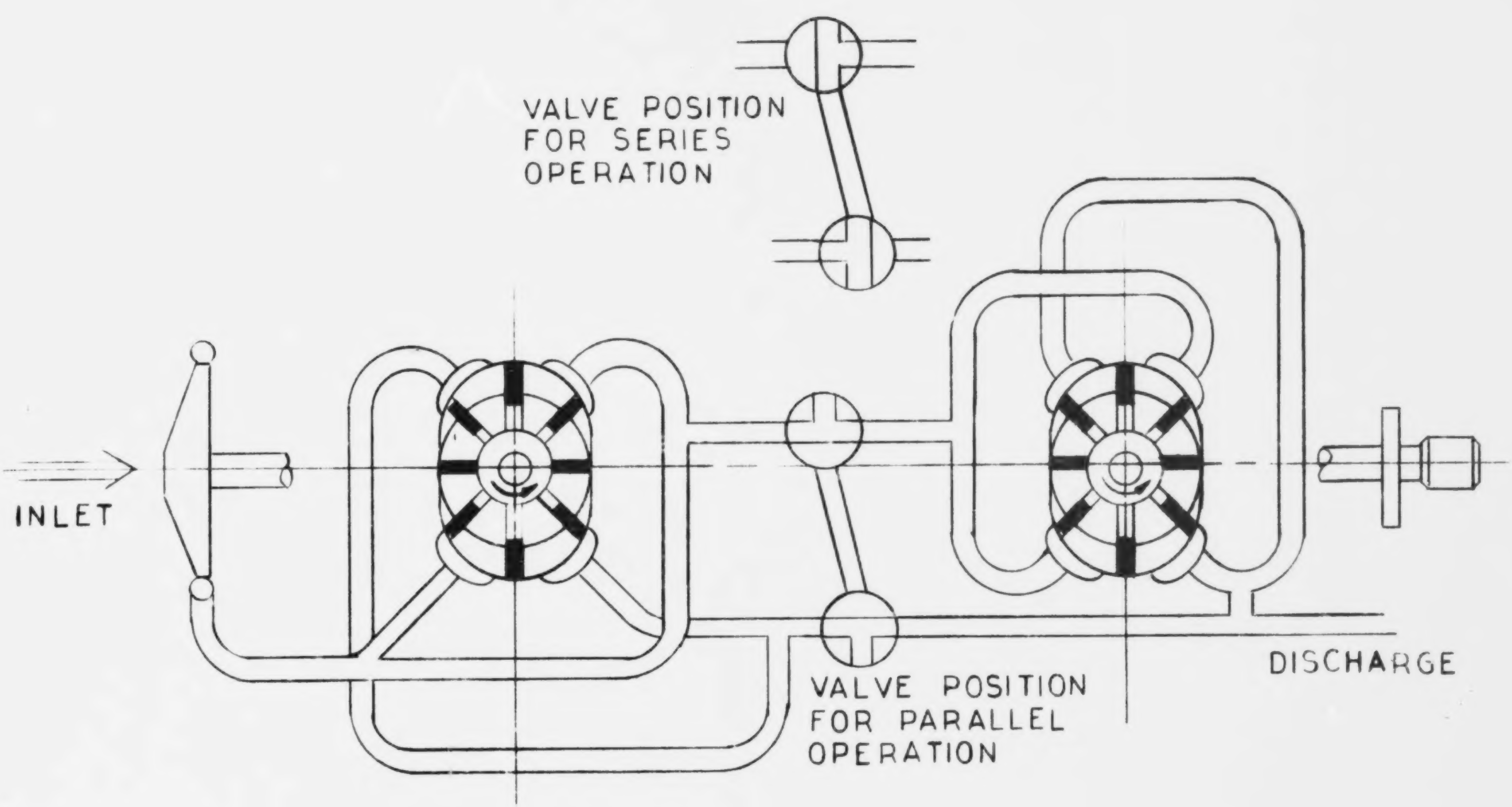
PREPARED BY RMR

VANE PUMP ON VARIABLE
SPEED DRIVE
ORGANIC FLUID
RECIPROCATOR (TECO)

CURVE TITLE

SYSTEM EFFICIENCY

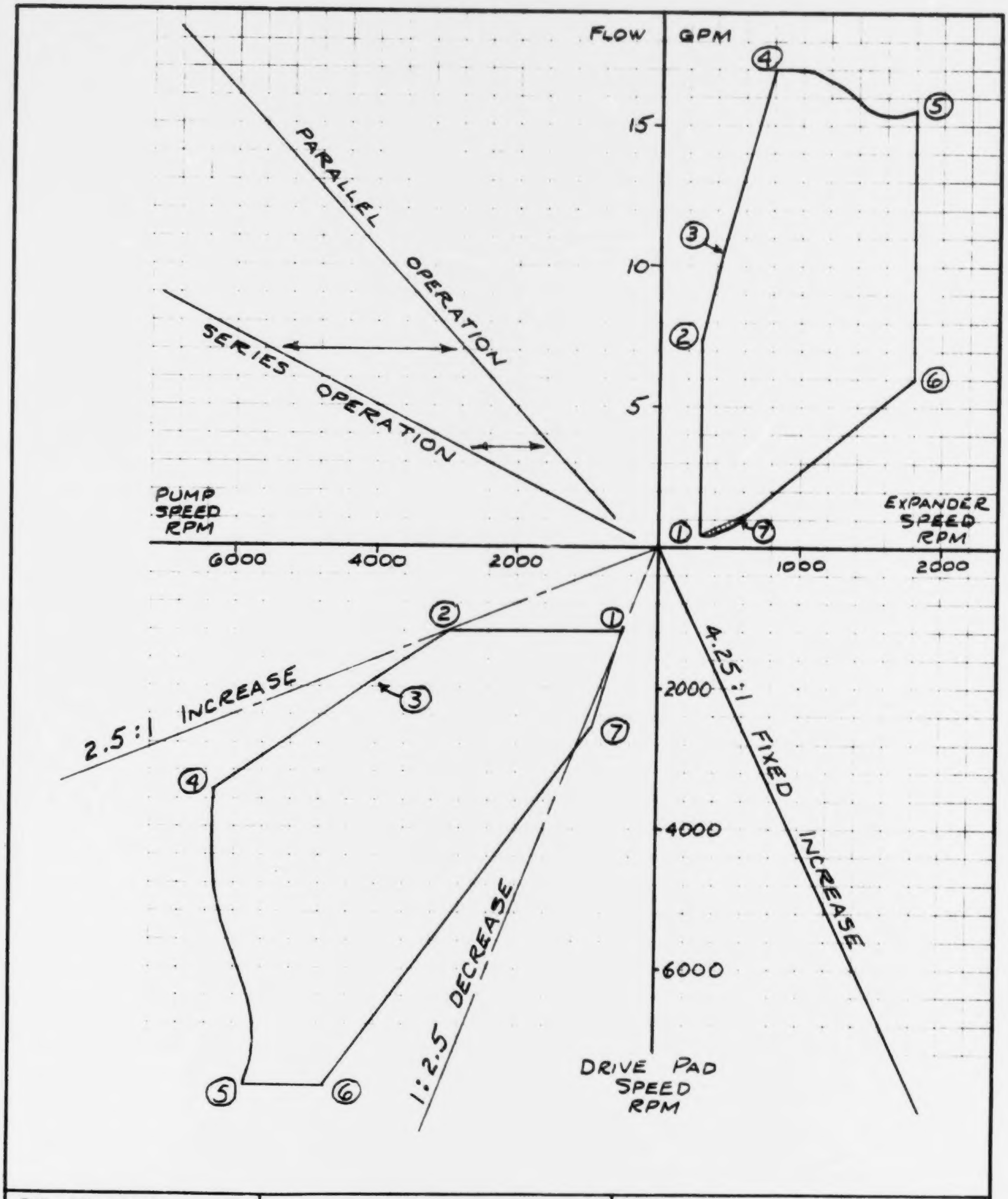
FIGURE NO. 9



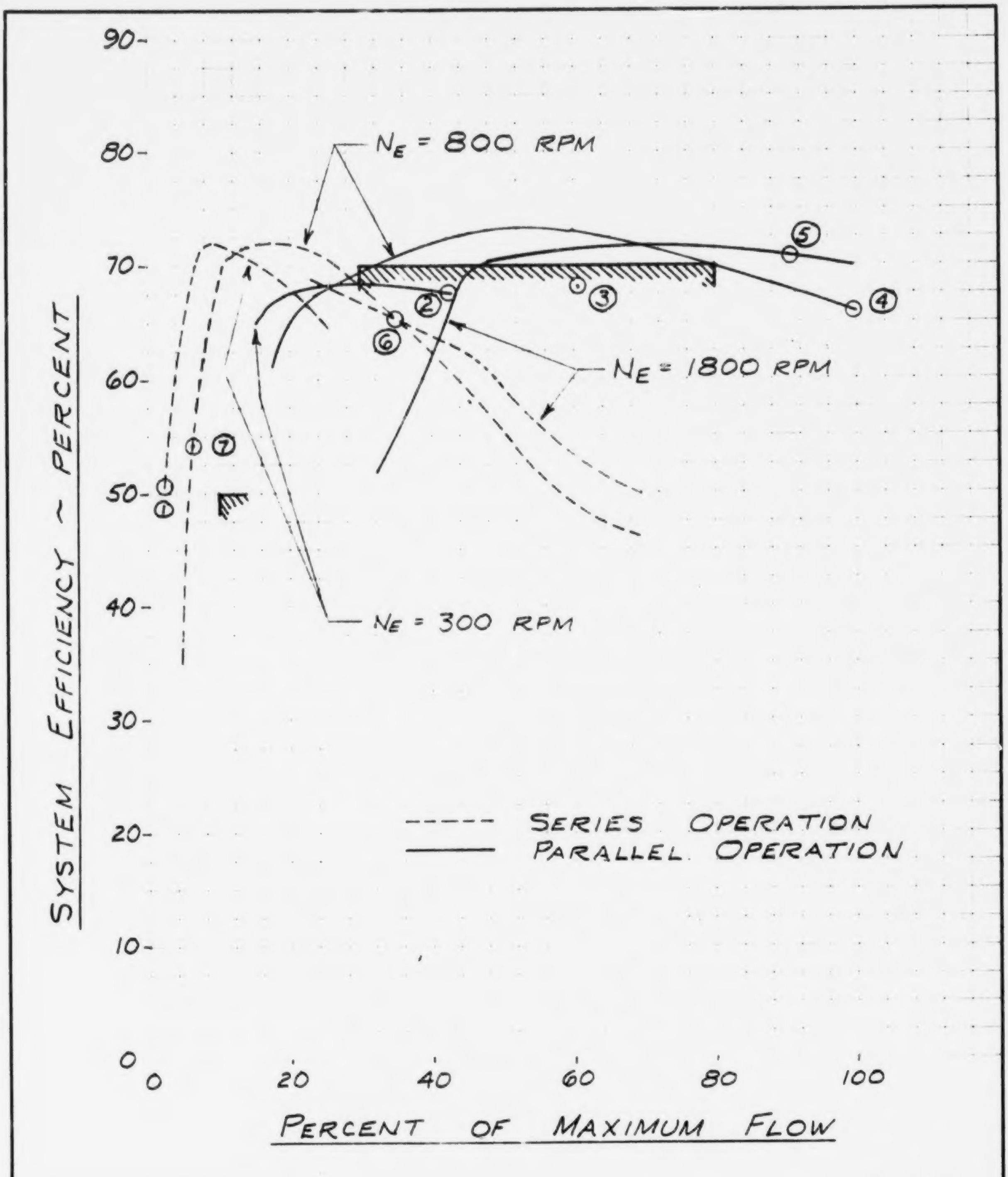
TWO STAGE VANE PUMP
SERIES - PARALLEL OPERATION

REPORT NO. APTD-1357

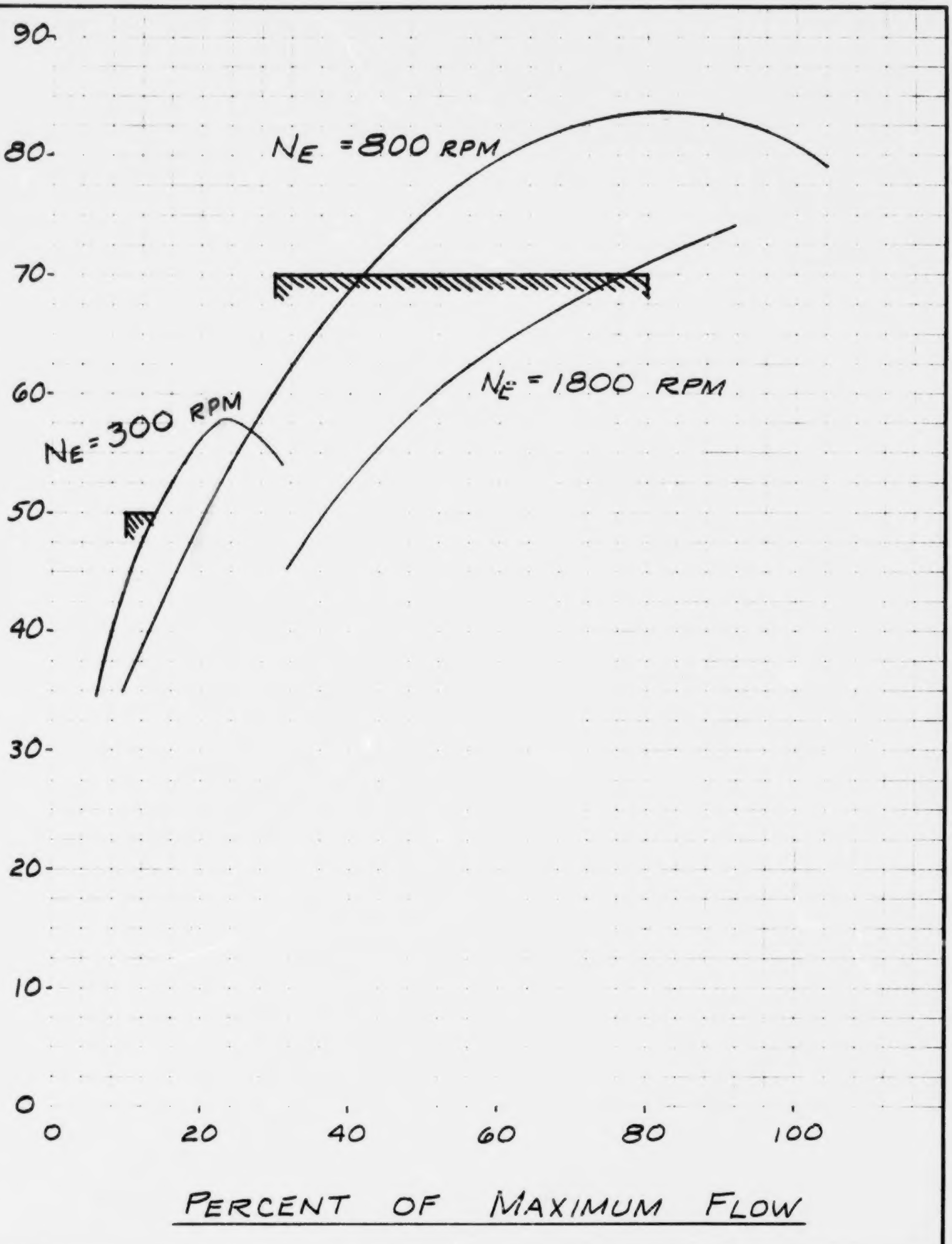
FIGURE NO. 10



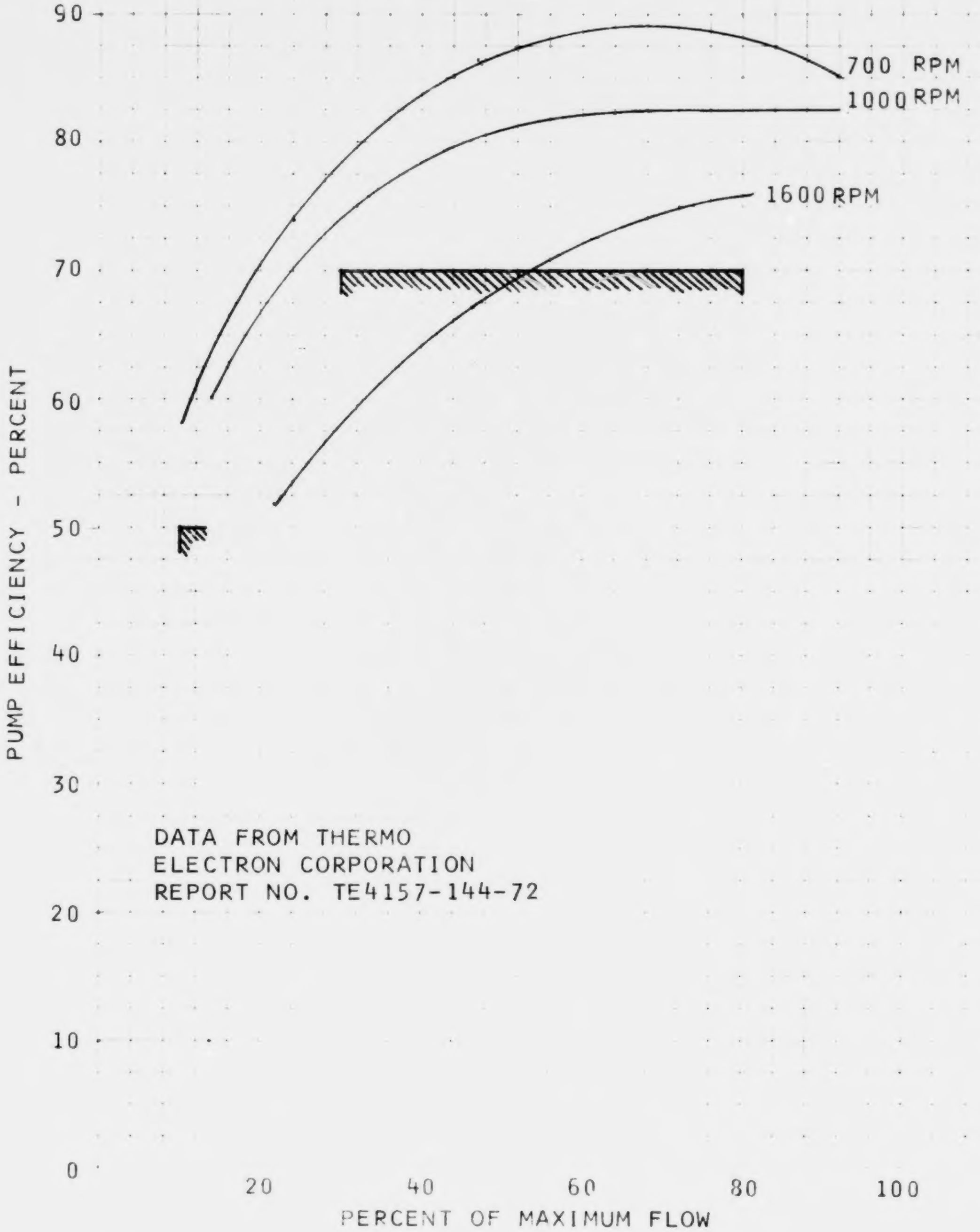
ENG. REPORT NO. APTD-1357	TWO STAGE VANE PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID RECIPROCATOR (TECO)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 11



ENG. REPORT NO. APTD-1357	TWO STAGE VANE PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID RECIPROCATOR (TECO)	CURVE TITLE SYSTEM EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 12

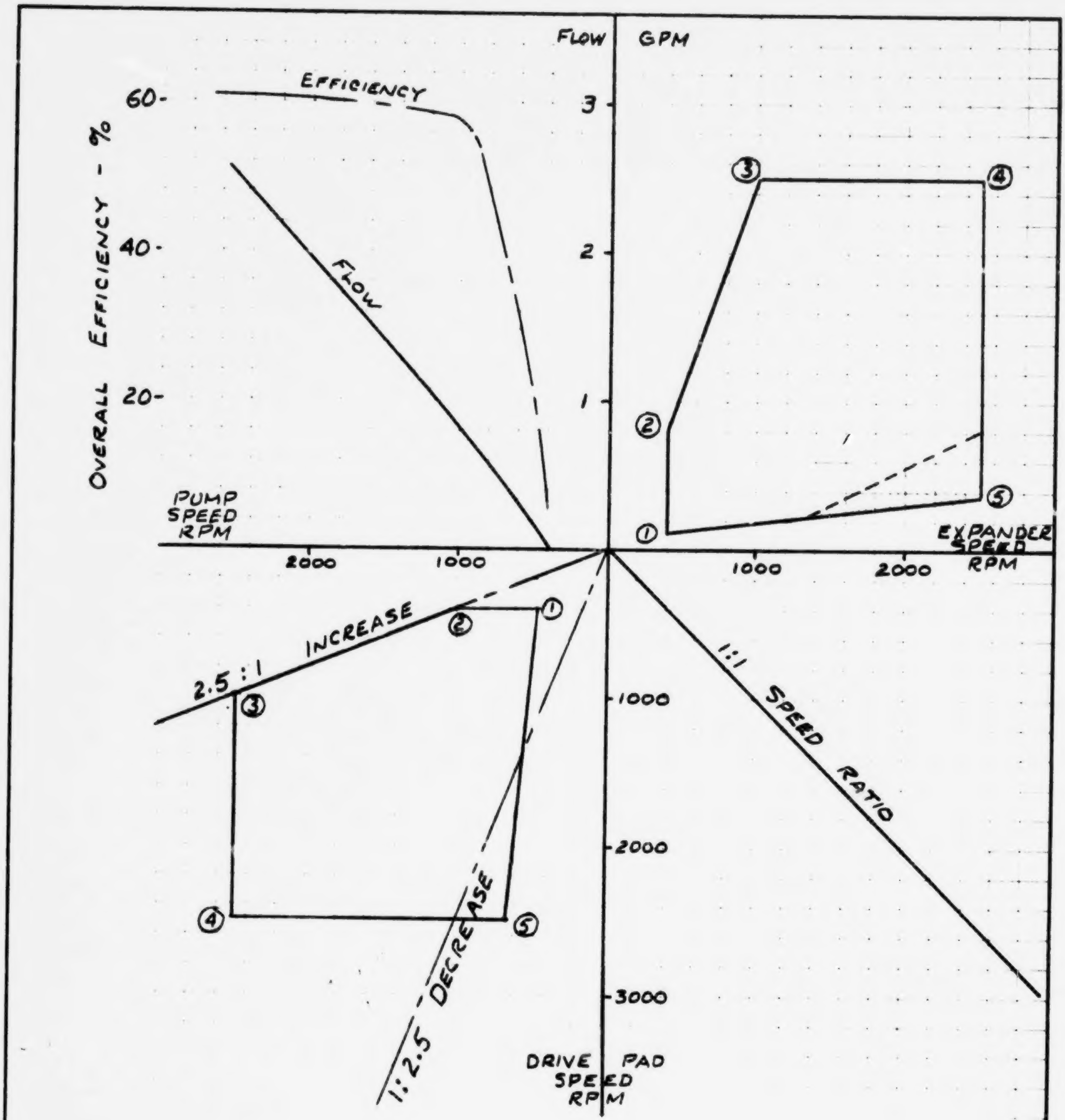


ENG. REPORT NO. APTD-1357	VARIABLE DISPLACEMENT VANE PUMP ORGANIC FLUID RECIPROCATOR (TECO)	**CURVE TITLE** PUMP EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 13

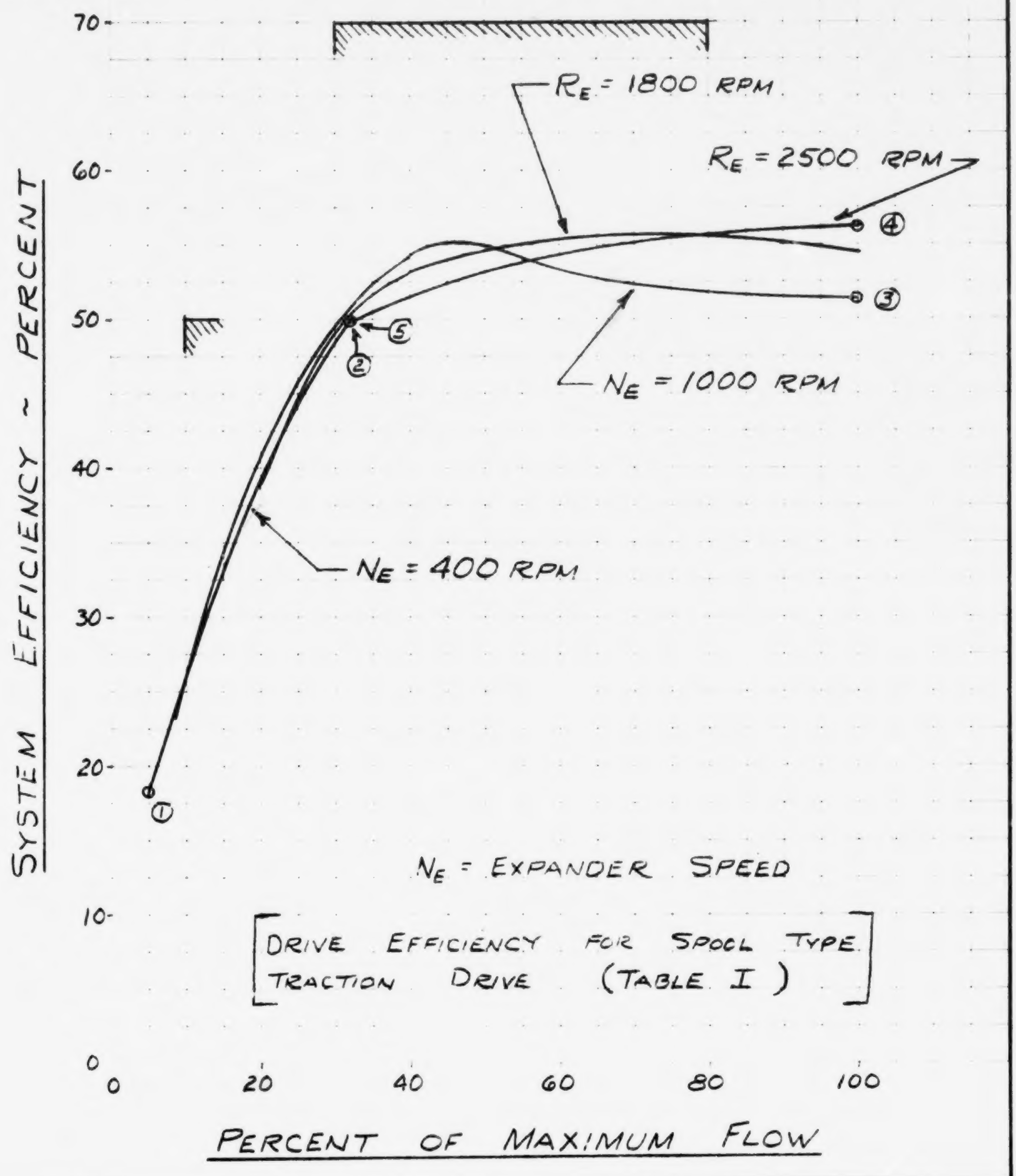


DATA FROM THERMO
ELECTRON CORPORATION
REPORT NO. TE4157-144-72

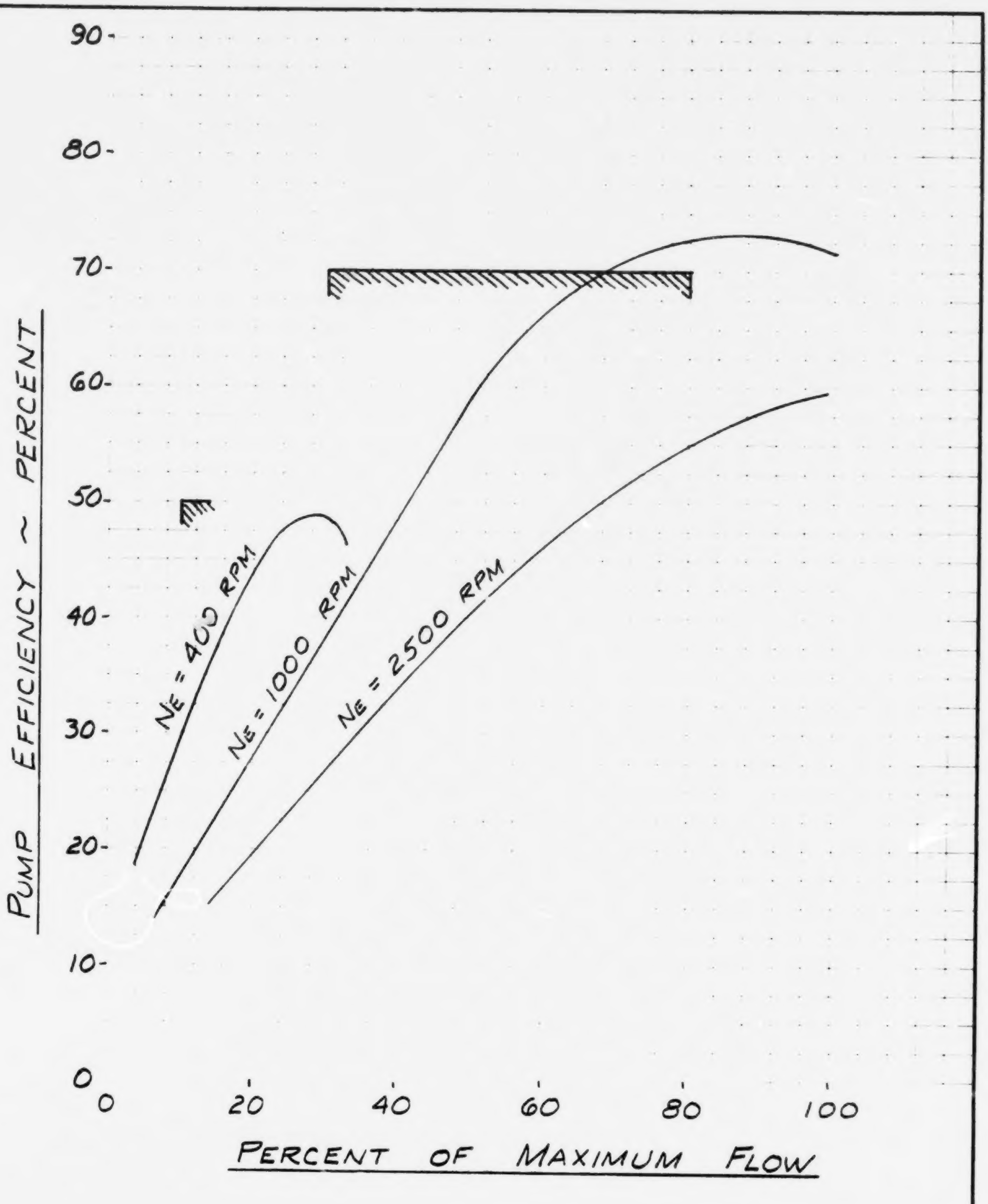
ENG. REPORT NO. APTD-1357	VARIABLE DISPLACEMENT PISTON PUMP ORGANIC FLUID RECIPROCATOR (TECO)	**CURVE TITLE** PUMP EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 14



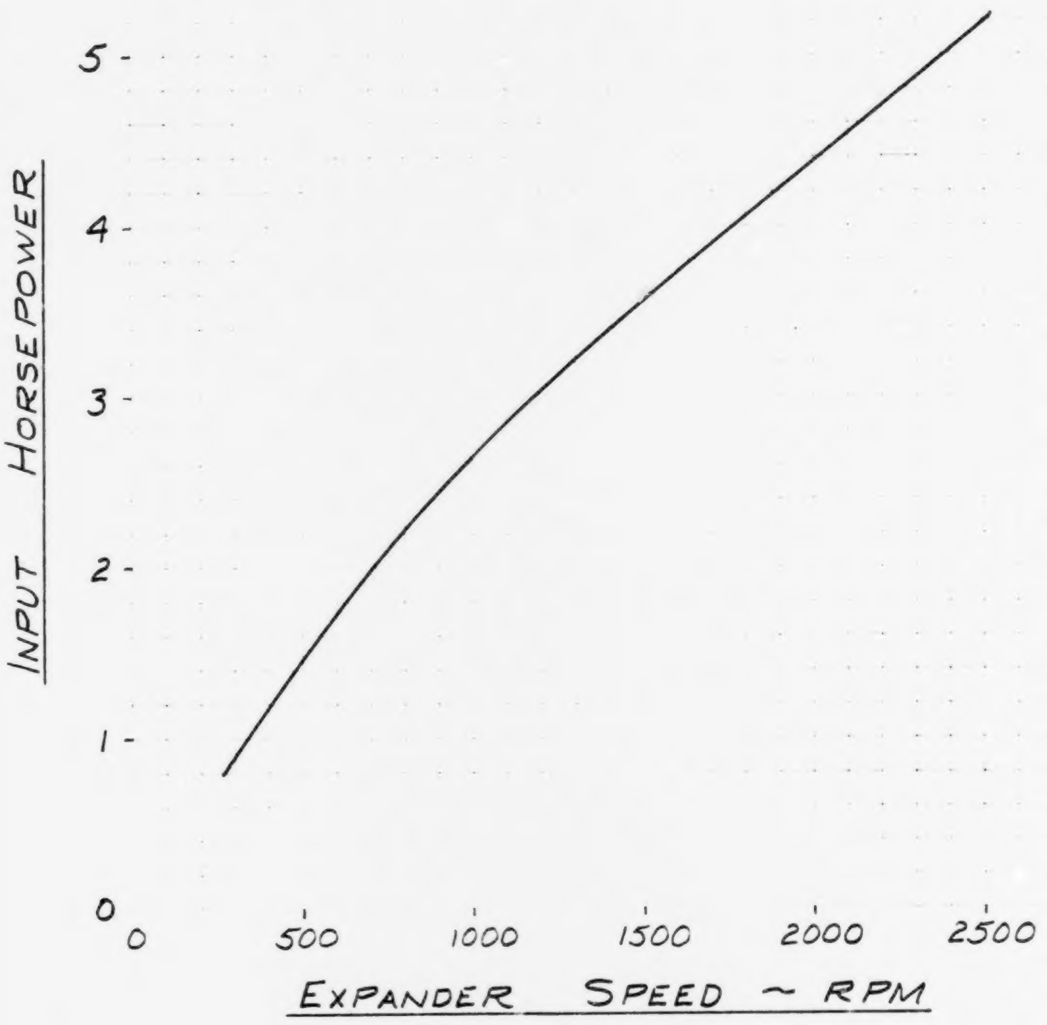
ENG. REPORT NO. APTD-1357	VANE PUMP ON VARIABLE SPEED DRIVE WATER RECIPROCATOR (SES)	**CURVE TITLE** PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 15



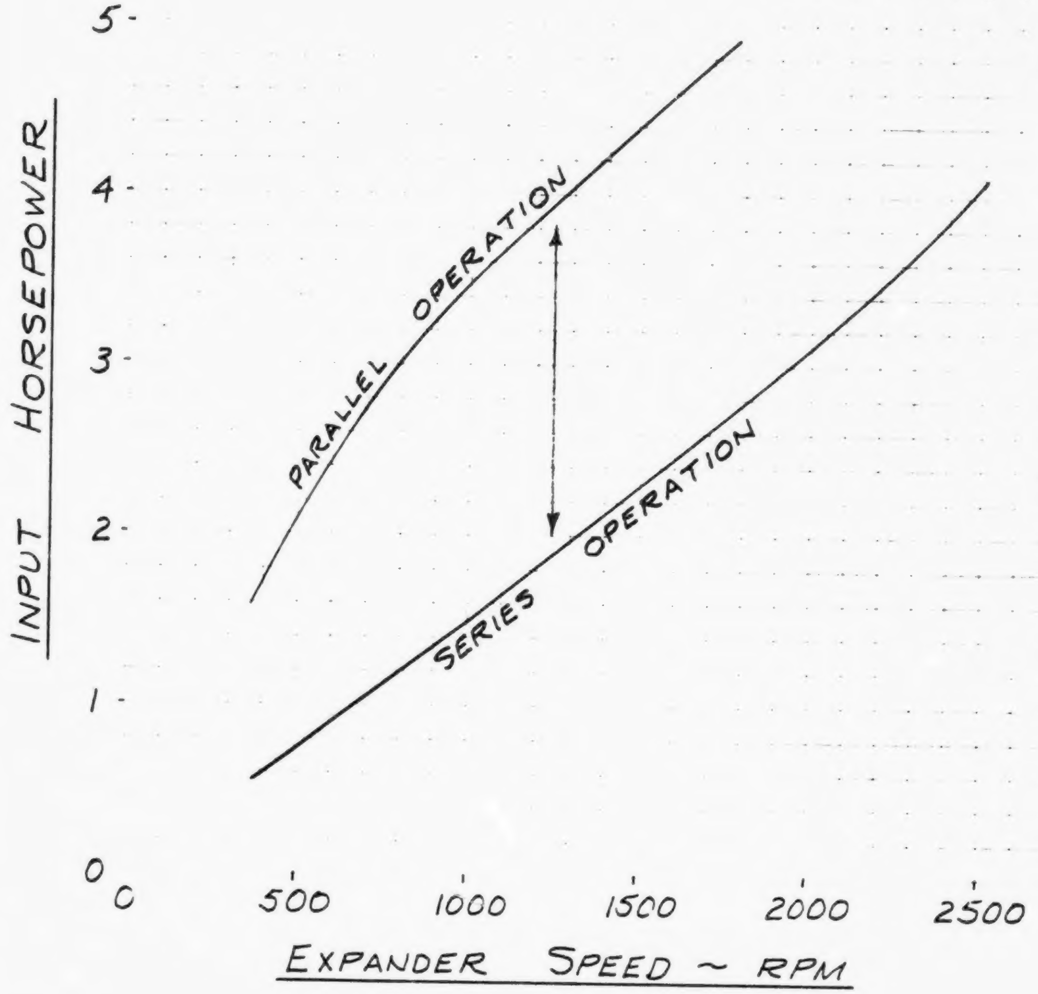
ENG. REPORT NO. APTD-1357	VANE PUMP ON VARIABLE SPEED DRIVE WATER RECIPROCATOR (SES)	**CURVE TITLE** SYSTEM EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 16



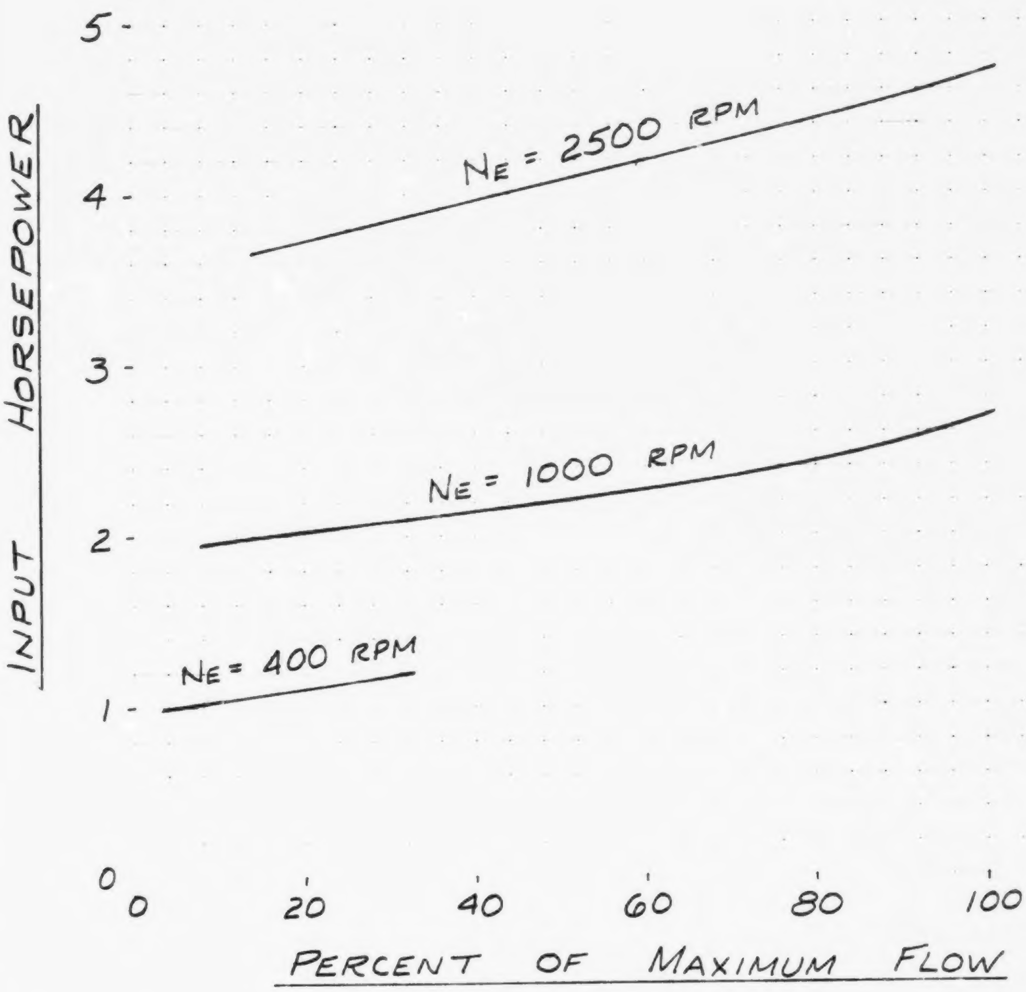
ENG. REPORT NO. APTD-1357	VARIABLE DISPLACEMENT VANE PUMP WATER RECIPROCATOR (SES)	CURVE TITLE PUMP EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 17



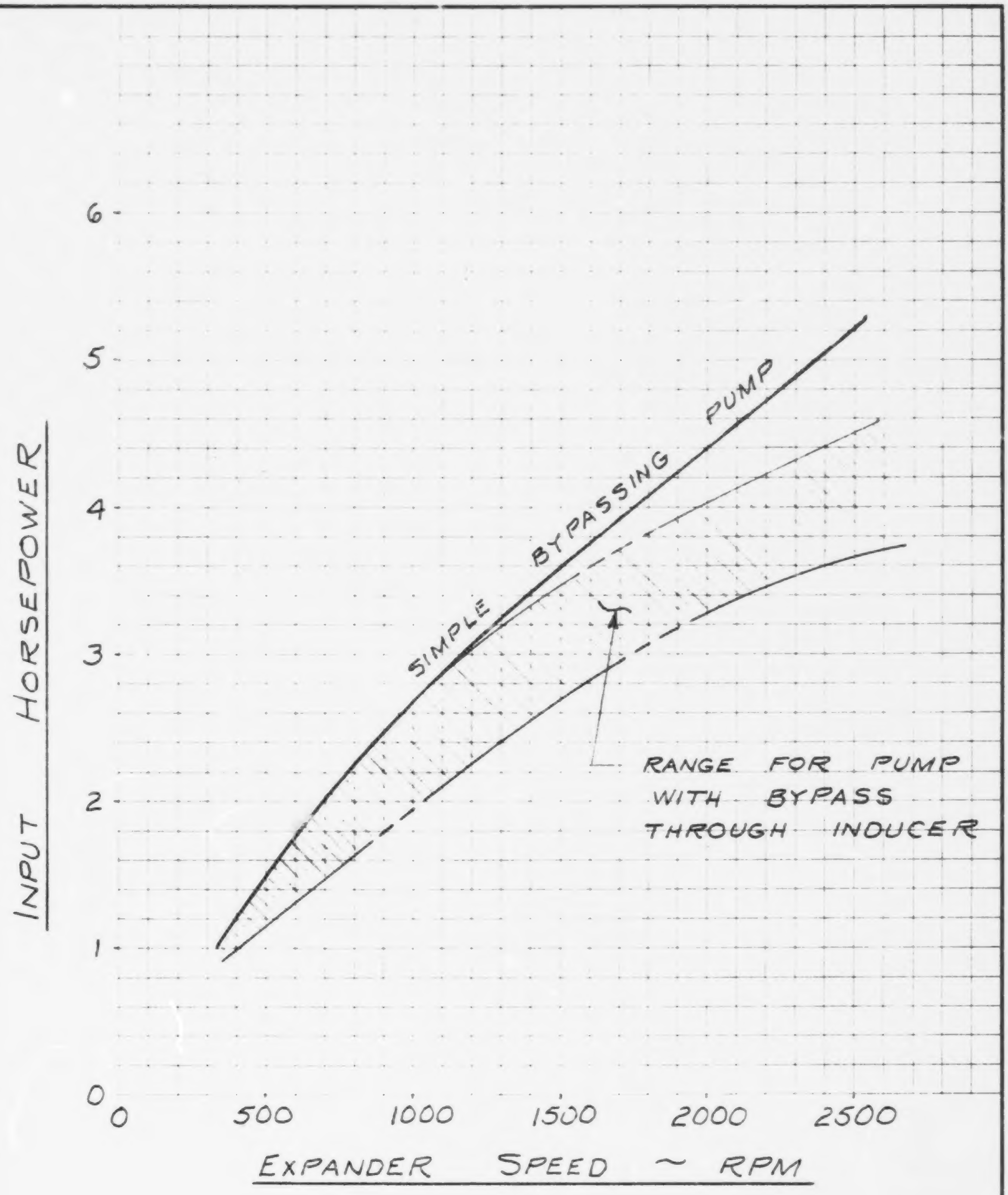
ENG. REPORT NO. APTD-1357	FIXED DISPLACEMENT VANE PUMP WITH BYPASS — WATER RECIPROCATOR (SES)	CURVE TITLE INPUT HORSEPOWER
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 18



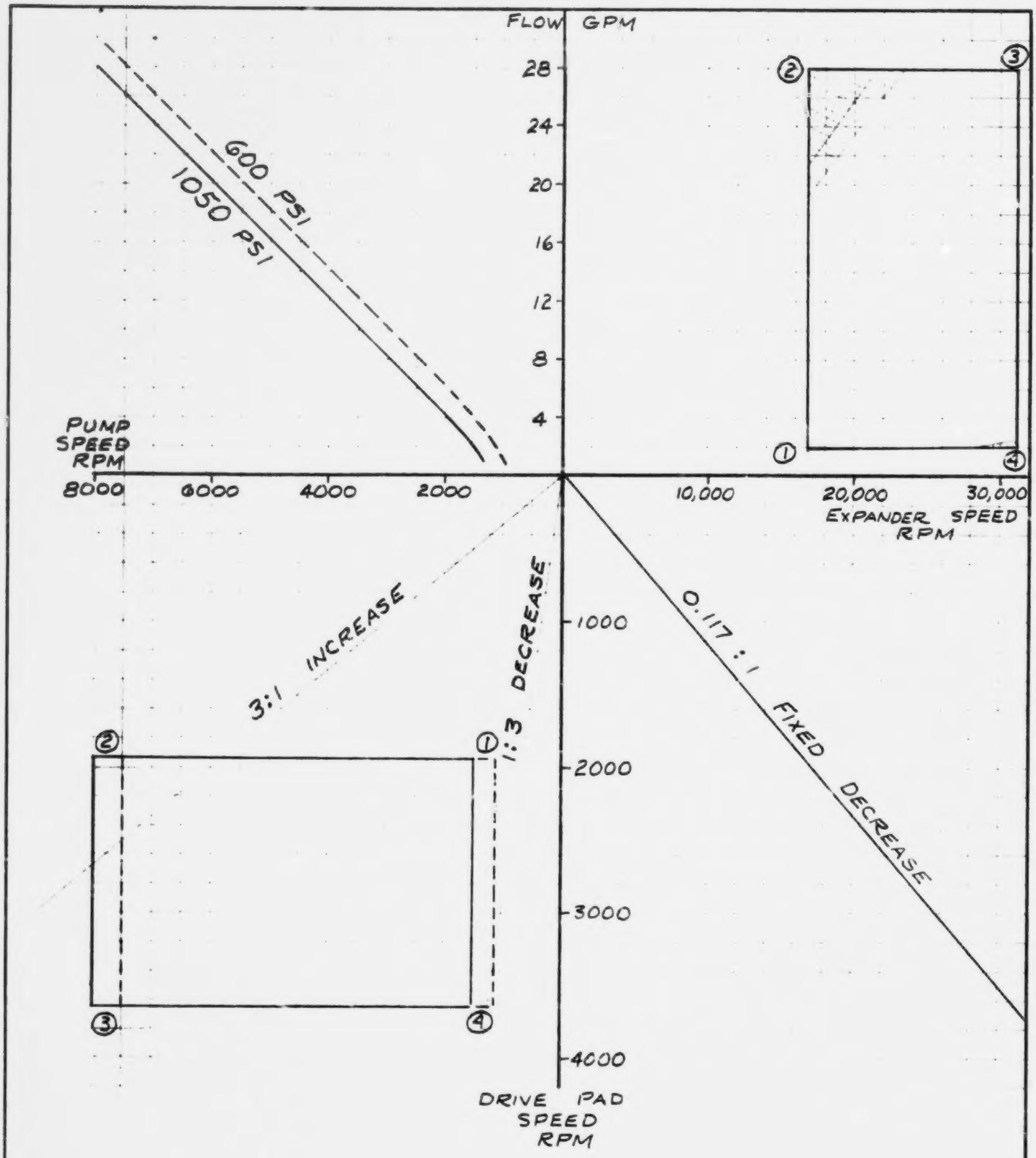
ENG. REPORT NO. APTD-1357	TWO - STAGE VANE PUMP WITH BYPASS — WATER RECIPROCATOR (SES)	**CURVE TITLE** INPUT HORSEPOWER
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 19



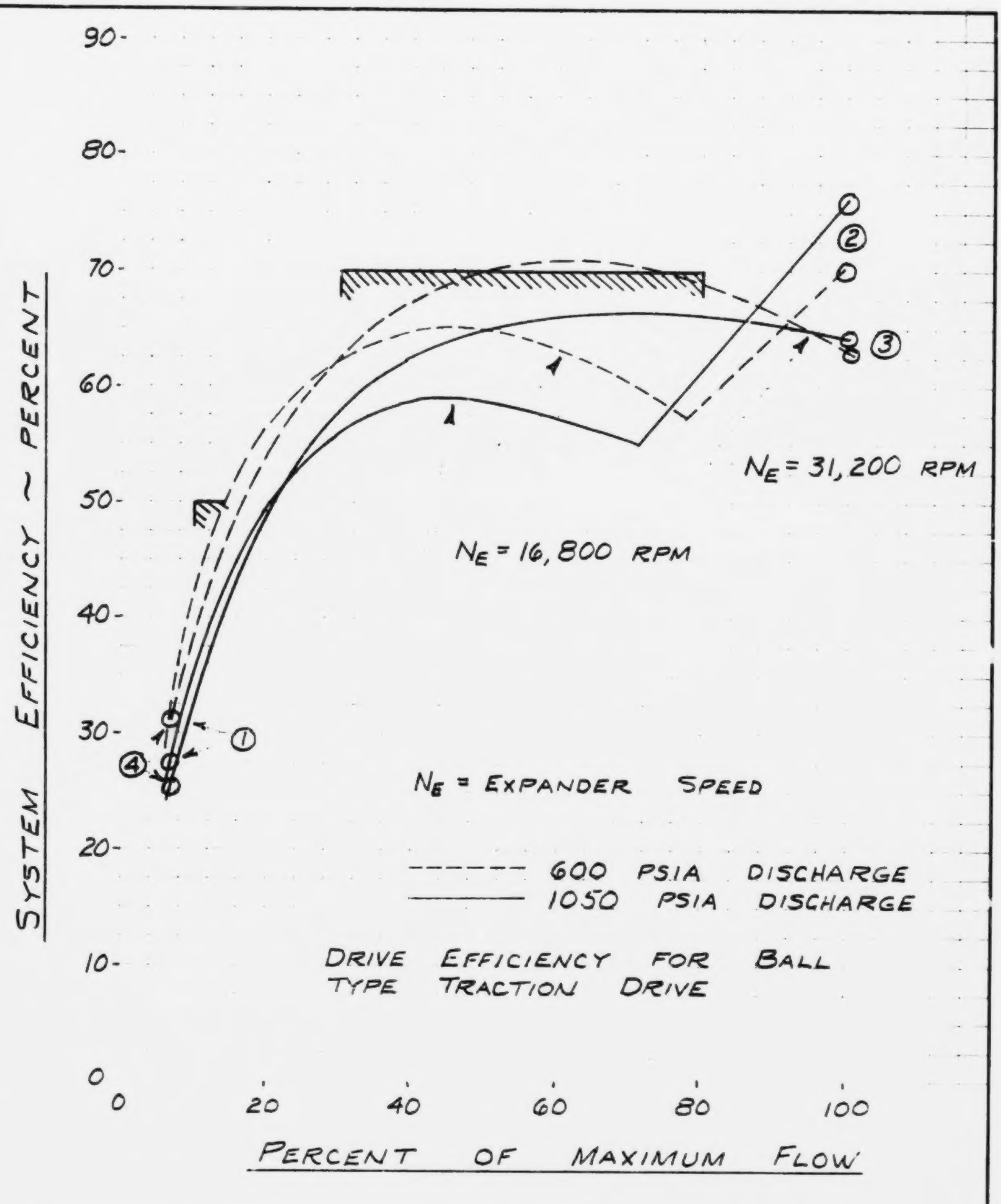
ENG. REPORT NO. APTD-1357	VANE PUMP WITH BYPASS THROUGH INDUCER — WATER RECIPROCATOR (SES)	**CURVE TITLE** INPUT HORSEPOWER
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 20



ENG. REPORT NO. APTD-1357	VANE PUMP WITH BYPASS THROUGH INDUCER — WATER RECIPROCATOR (SES)	CURVE TITLE INPUT HORSEPOWER COMPARED WITH SIMPLE BYPASSING PUMP
DATE 12-5-72		
PREPARED B. RMR		FIGURE NO. 27



ENG. REPORT NO. APTD-1357	GEAR PUMP ON VARIABLE SPEED DRIVE – ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 22



ENG. REPORT NO. APTD-1357

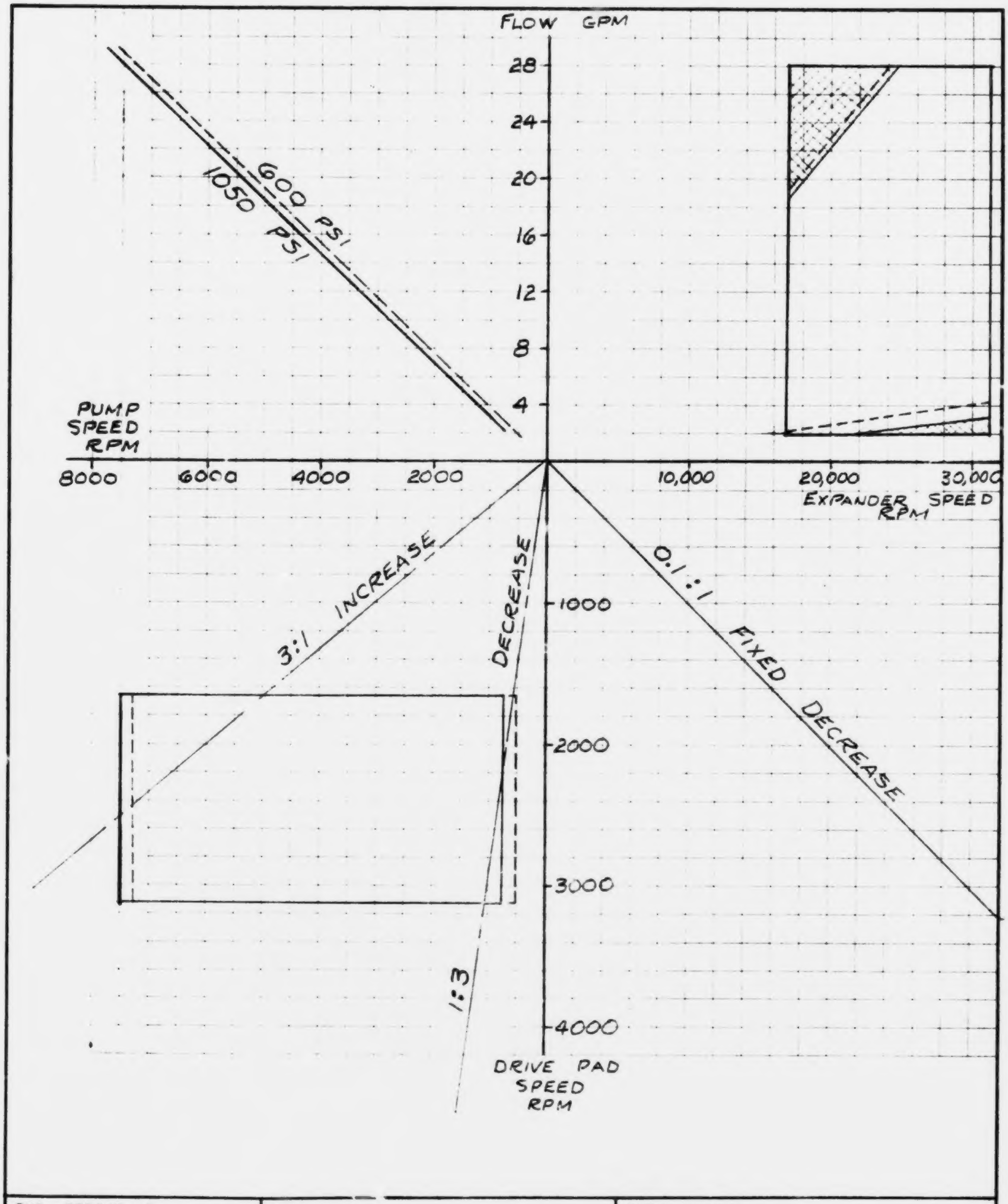
DATE 12-5-72

PREPARED BY RMR

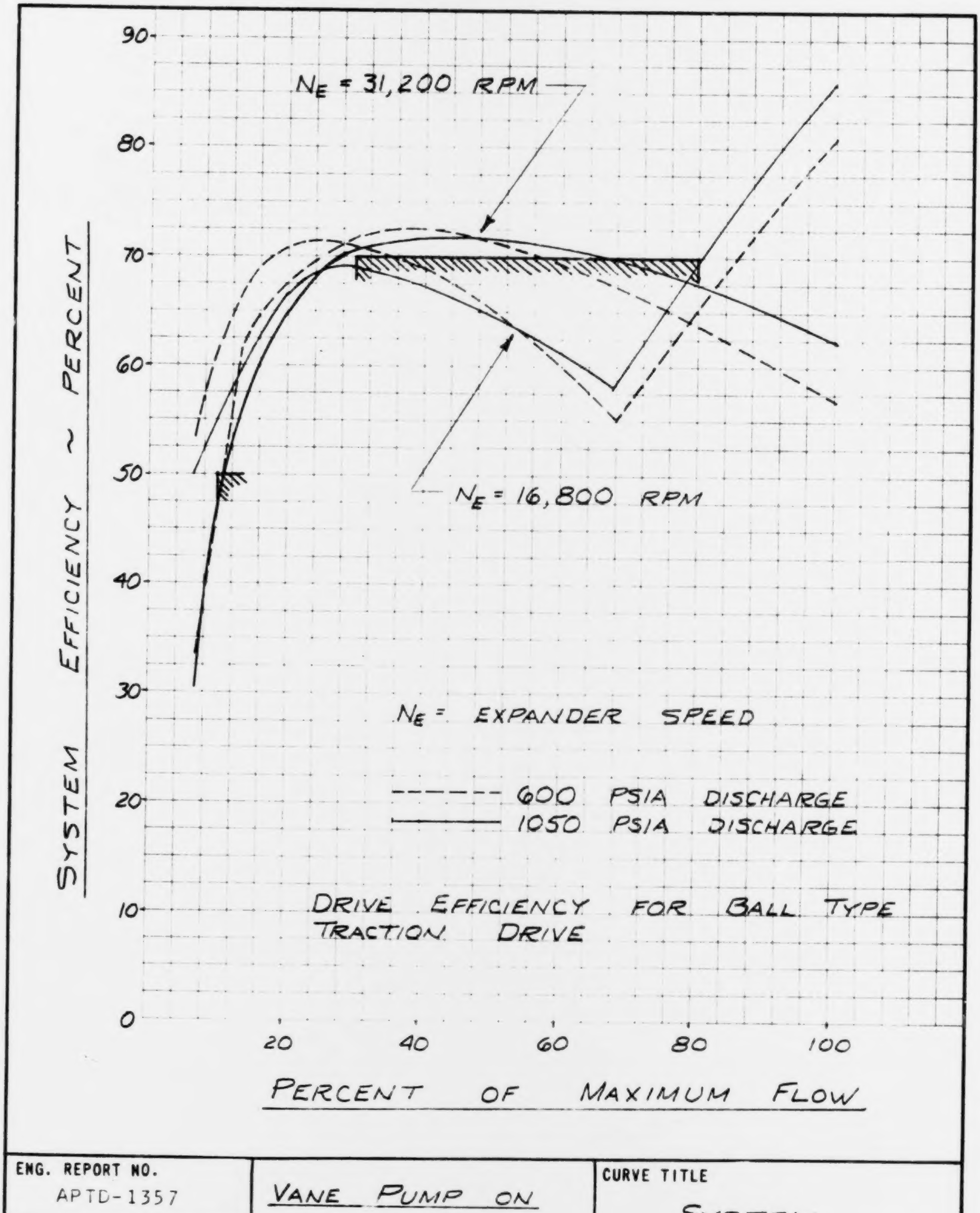
GEAR PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (ALRC)

CURVE TITLE SYSTEM EFFICIENCY

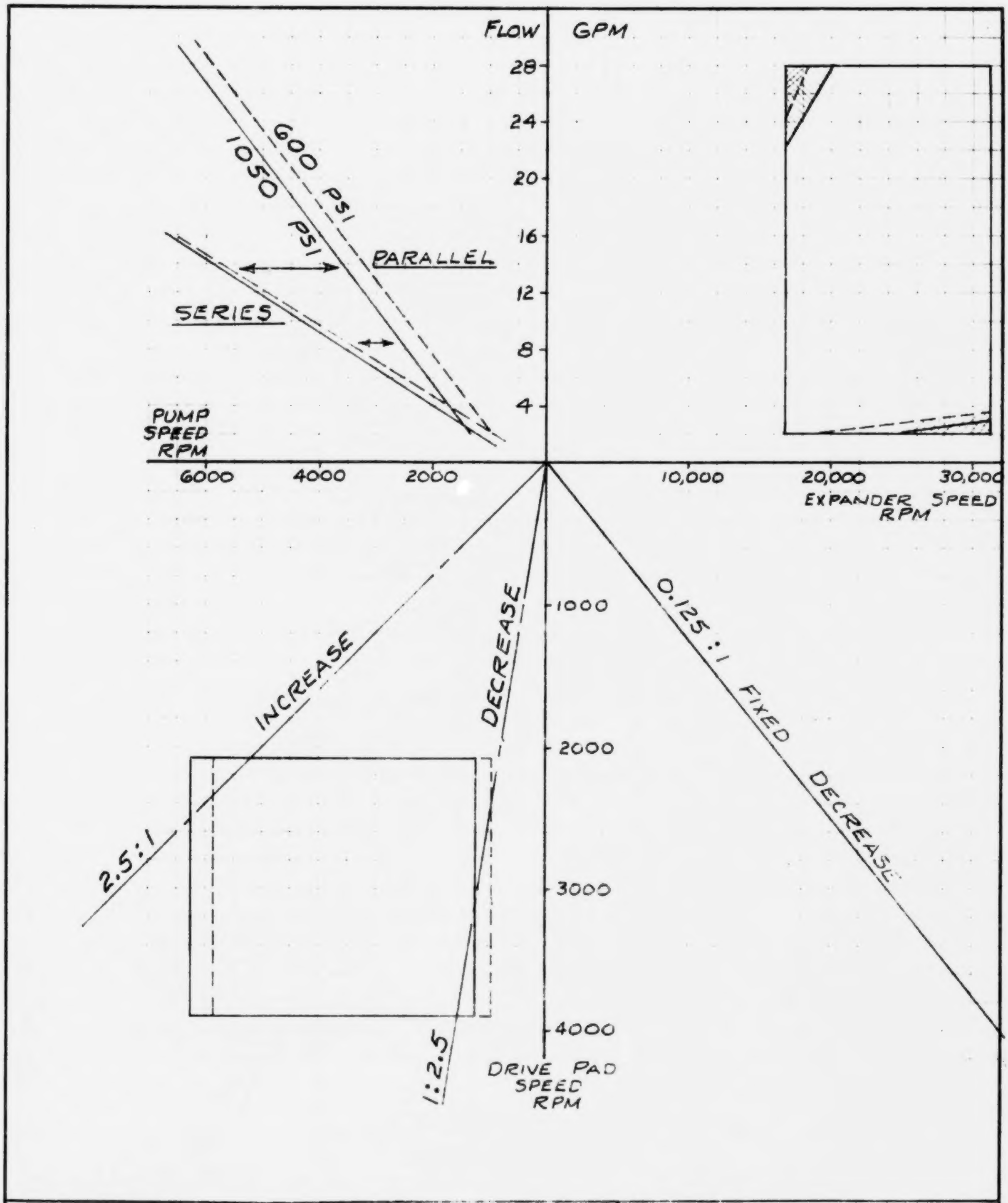
FIGURE NO. 23



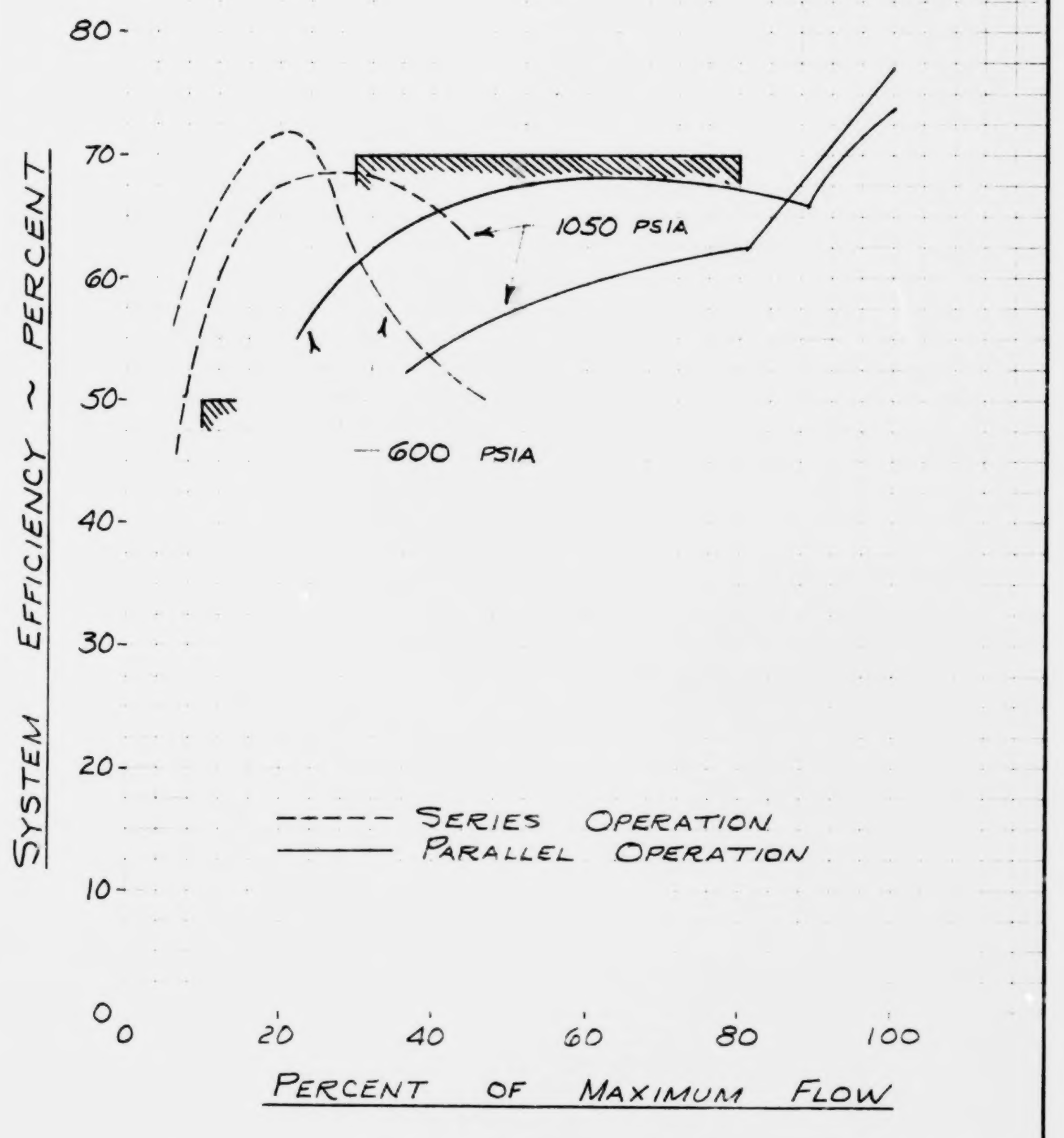
ENG. REPORT NO. APTD-1357	VANE PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 24



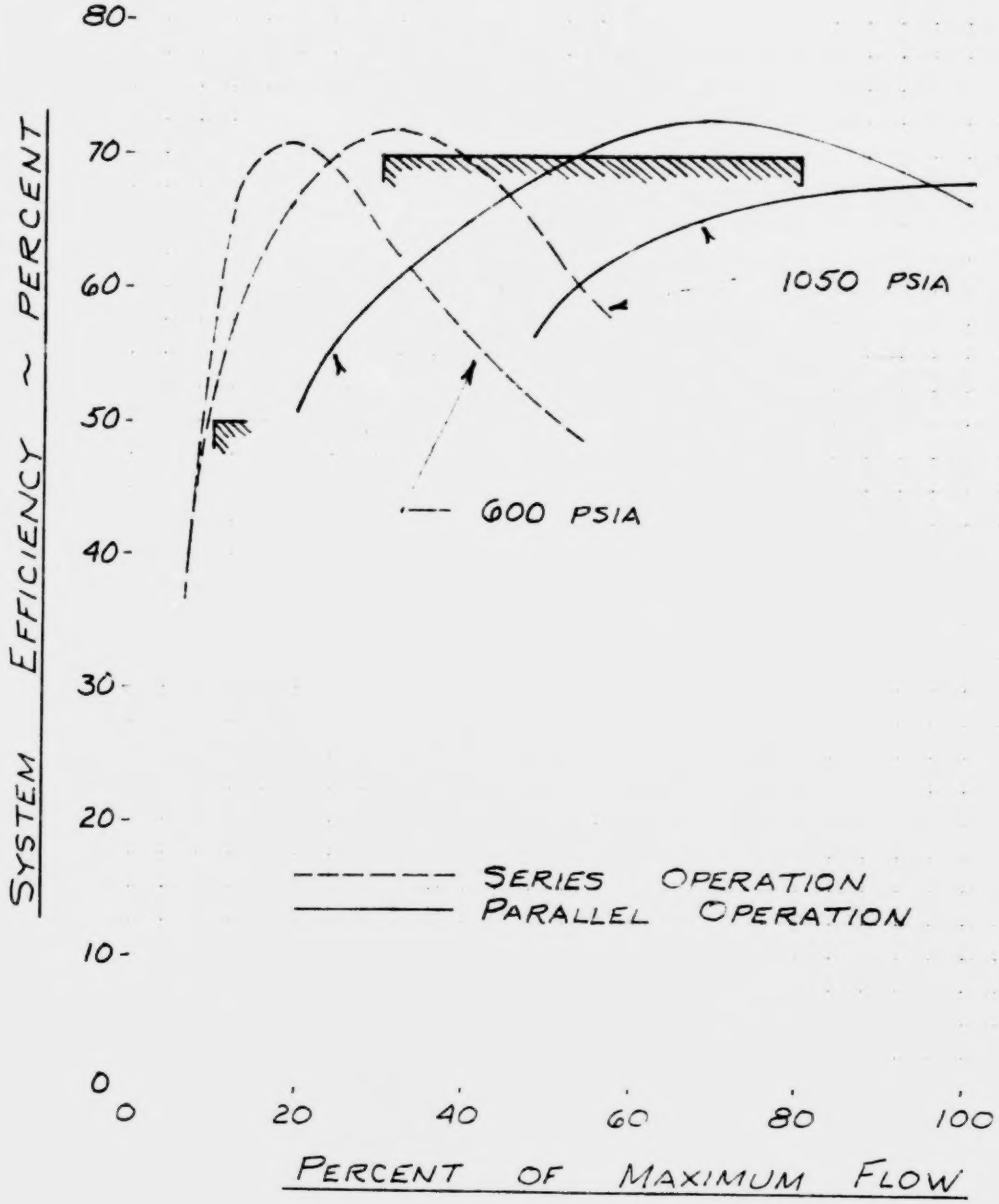
ENG. REPORT NO. APTD-1357	VANE PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE SYSTEM EFFICIENCY
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 25



ENG. REPORT NO. APTD-1357	TWO-STAGE GEAR PUMP ON VARIABLE SPEED DRIVE - ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE PERFORMANCE ENVELOPE
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 26



ENG. REPORT NO. APTD-1357	TWO-STAGE GEAR PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE SYSTEM EFFICIENCY AT 16,800 RPM
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 27



ENG. REPORT NO. APTD-1357

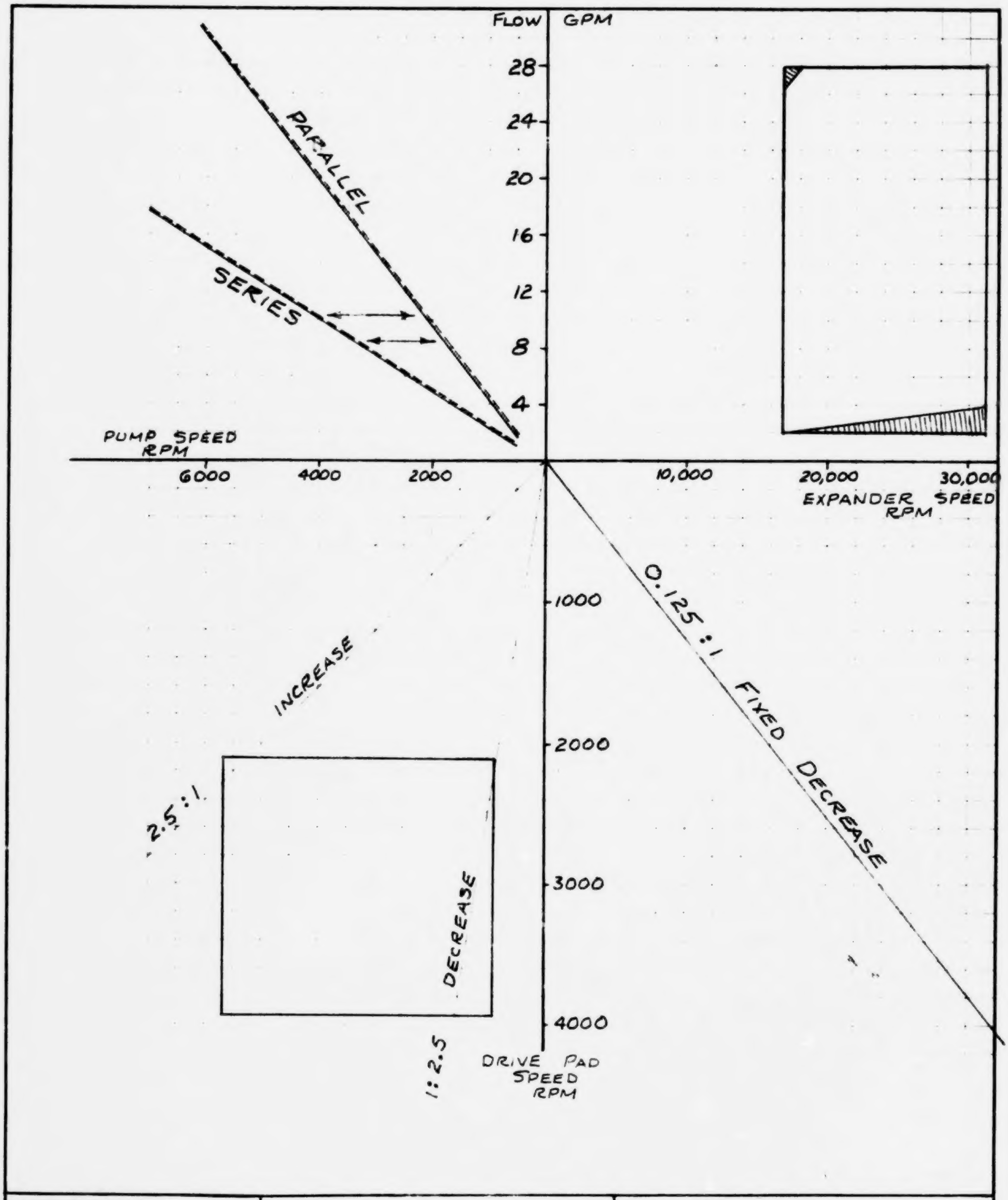
DATE 12-5-72

PREPARED BY RMR

TWO-STAGE GEAR PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (ALRC)

CURVE TITLE SYSTEM EFFICIENCY AT 31,200 RPM

FIGURE NO. 28



ENG. REPORT NO. APTD-1357

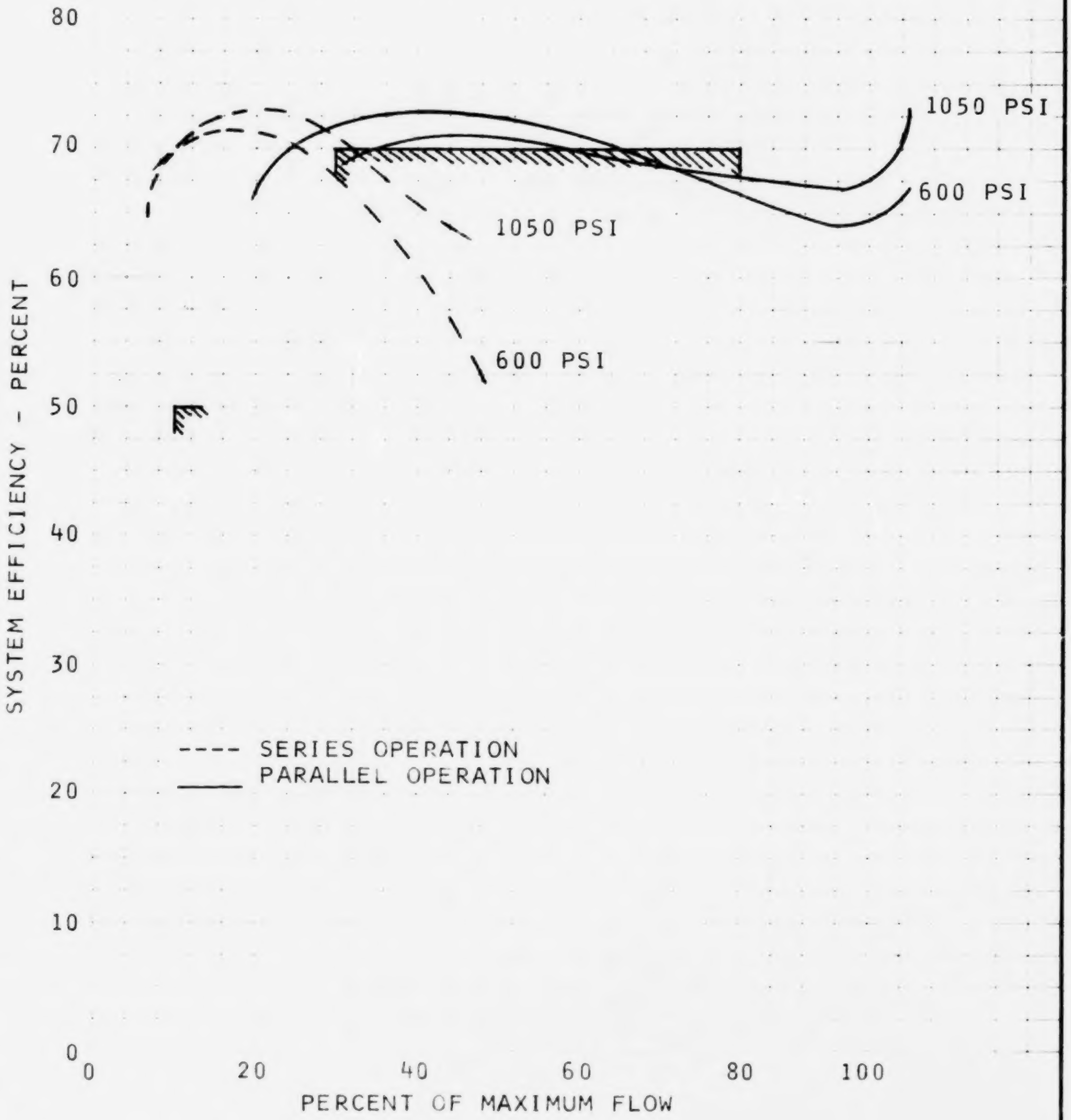
DATE 12-5-72

PREPARED BY RMR

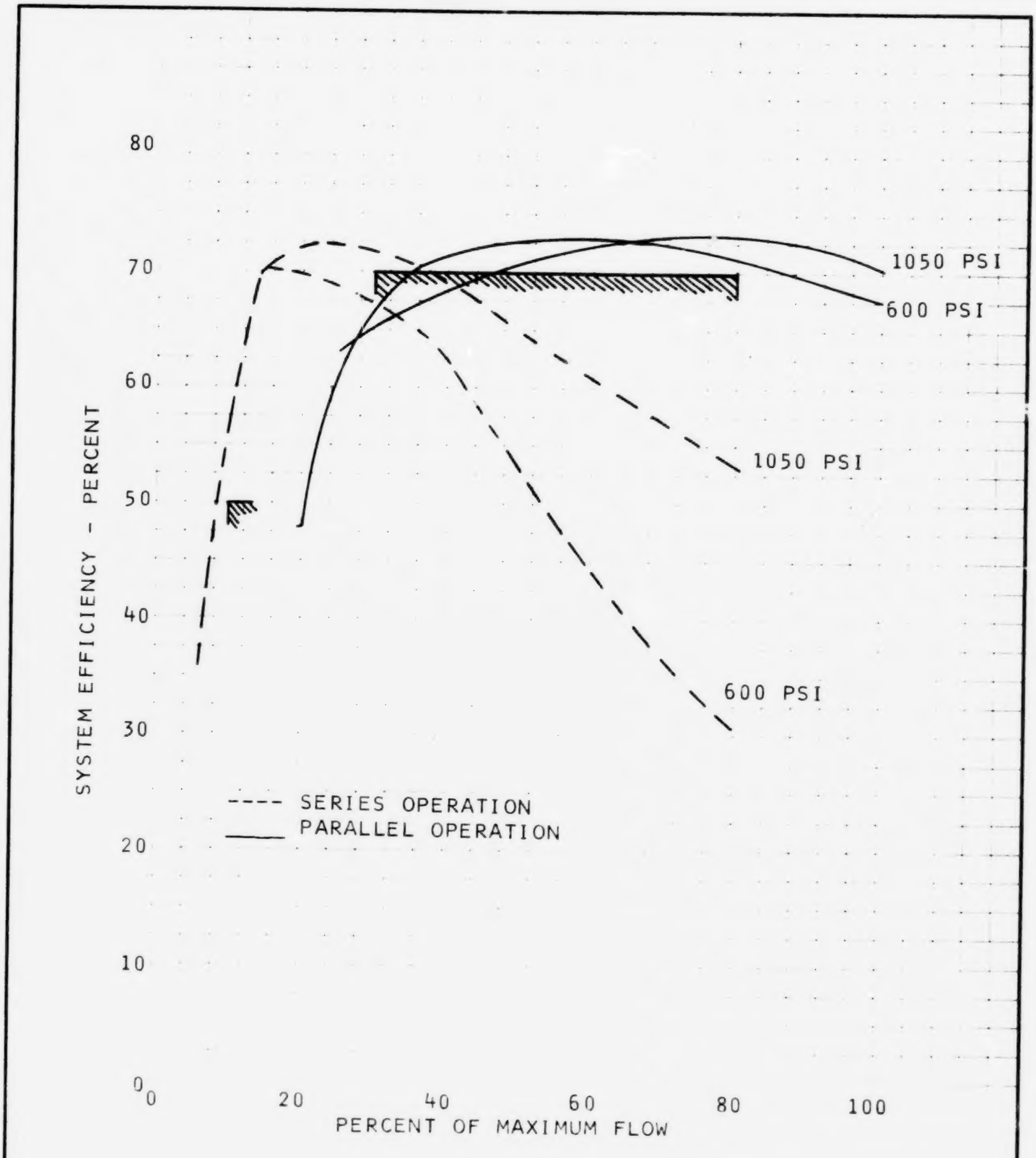
TWO-STAGE VANE PUMP ON VARIABLE SPEED DRIVE — ORGANIC FLUID TURBINE (AI.RC)

CURVE TITLE PERFORMANCE ENVELOPE

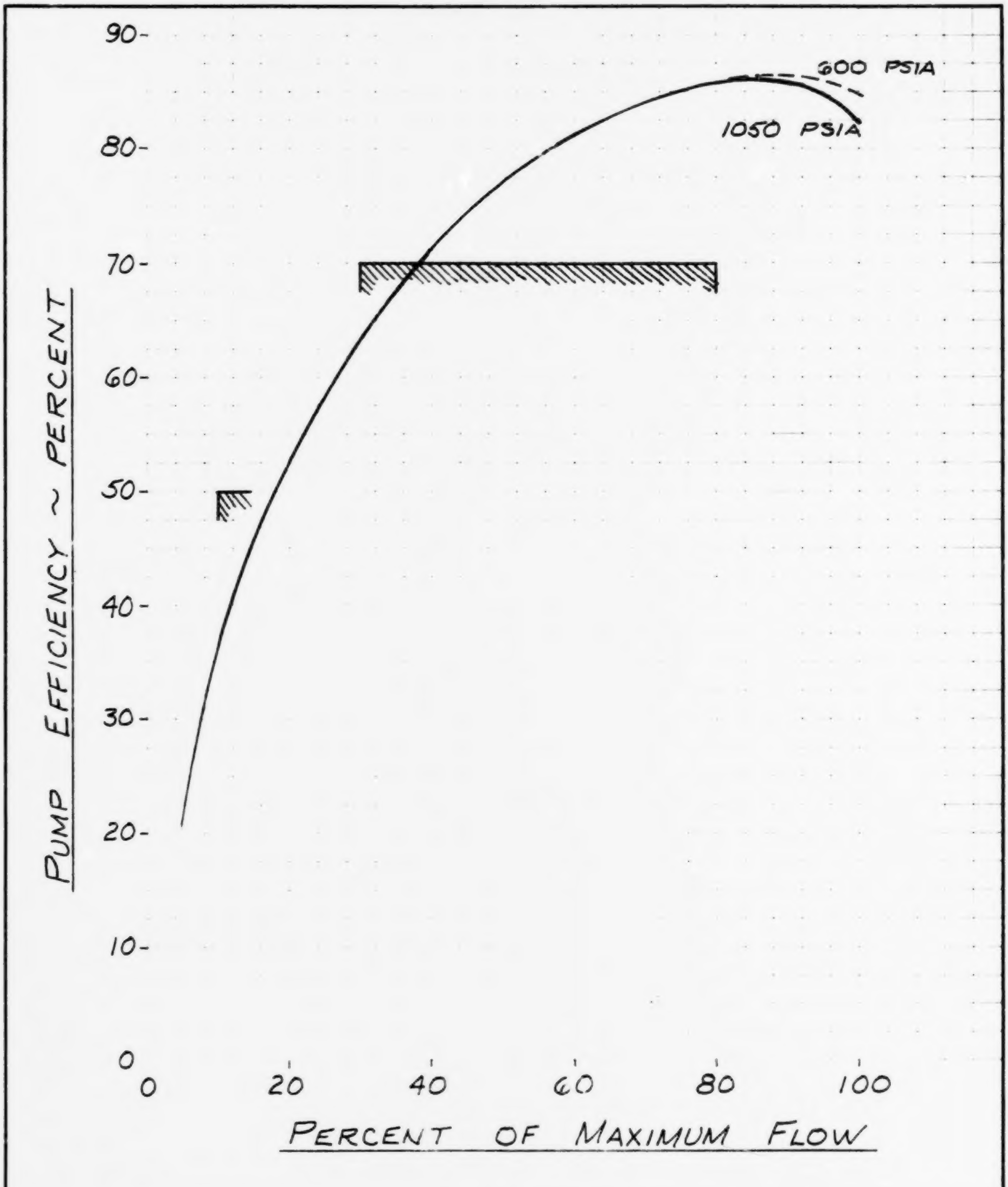
FIGURE NO. 29



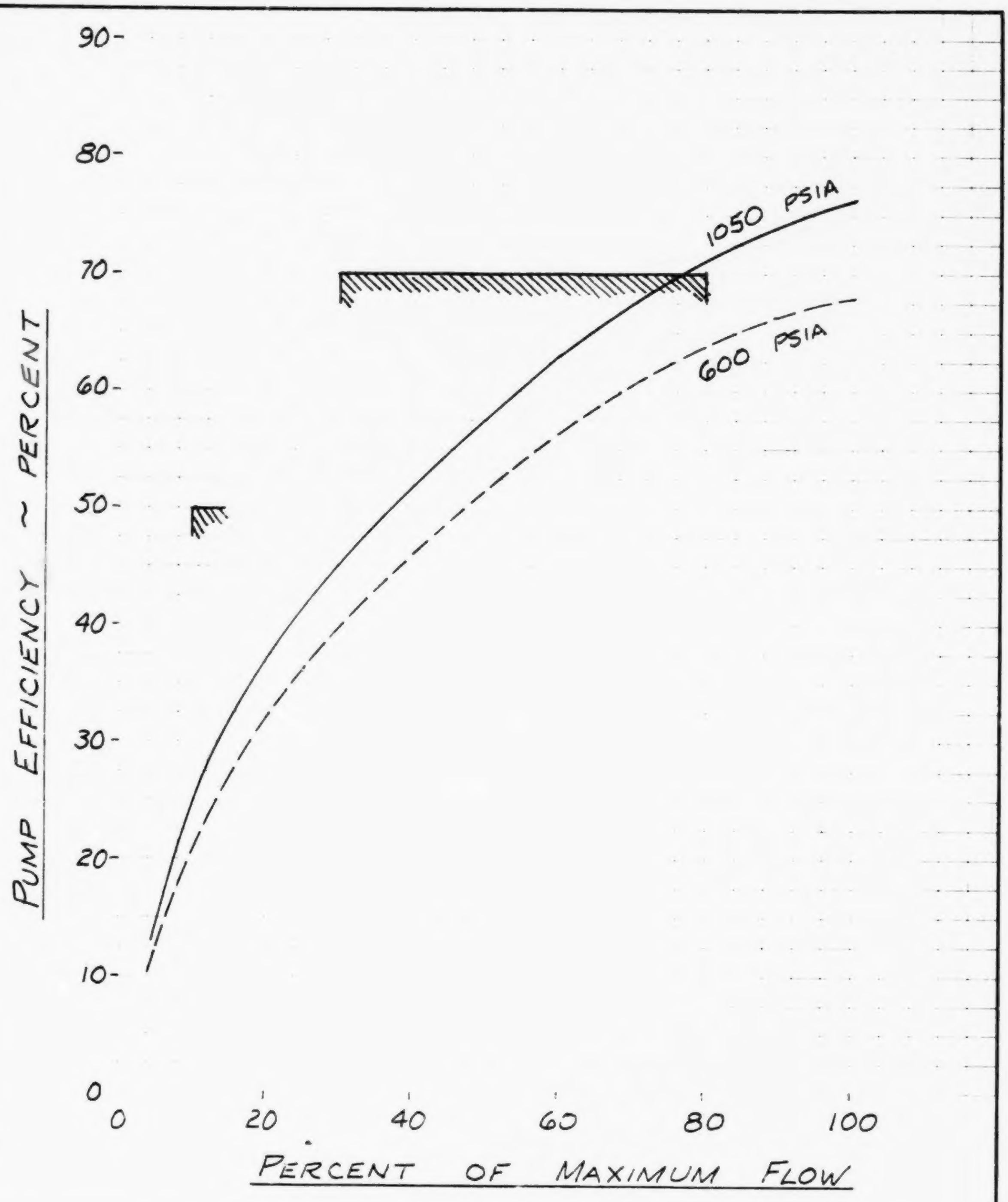
ENG. REPORT NO. APTD-1357	TWO STAGE VANE PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID TURBINE (ALRC)	**CURVE TITLE** SYSTEM EFFICIENCY AT 16,800 RPM
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 30



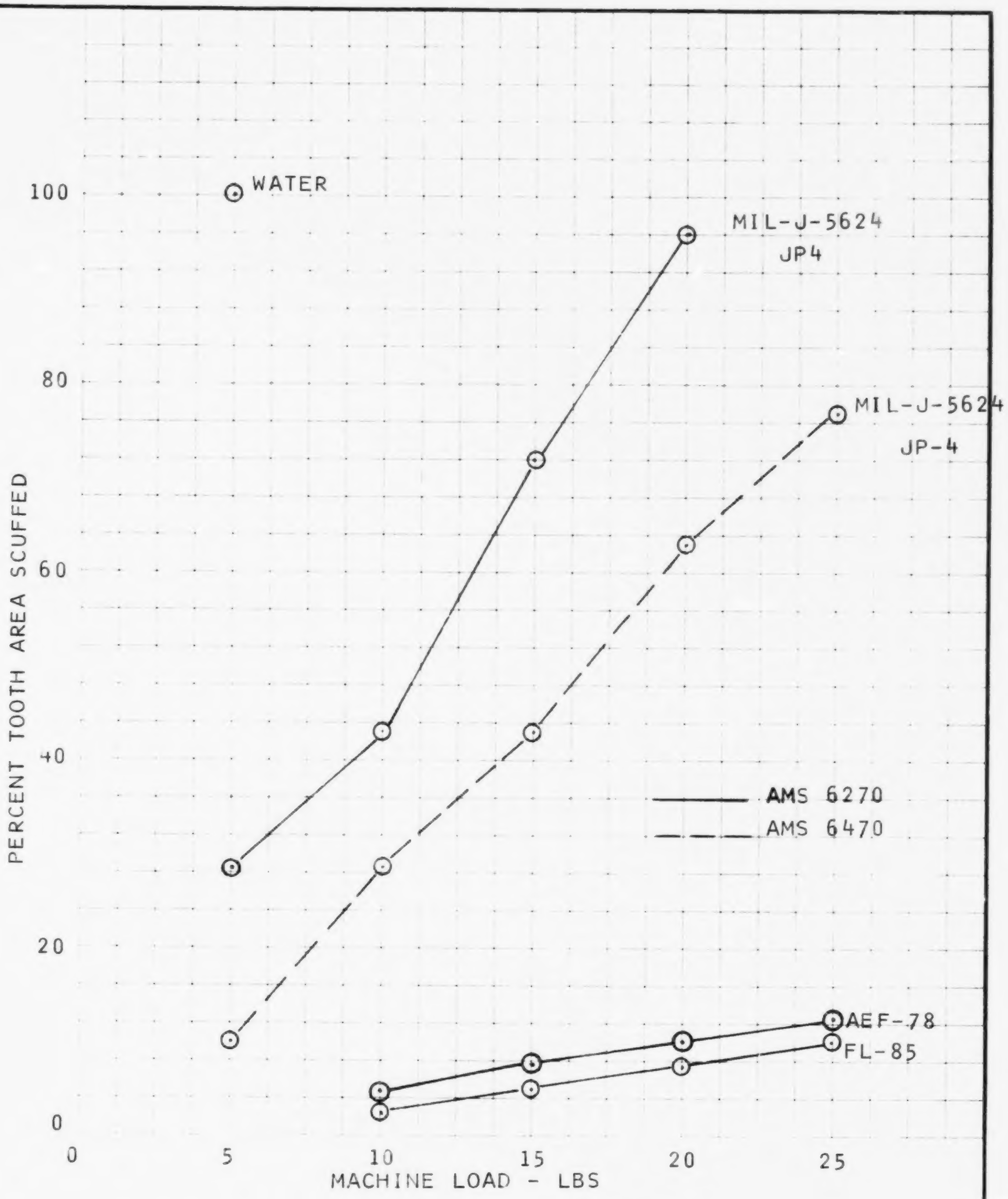
ENG. REPORT NO. APTD-1357	TWO STAGE VANE PUMP ON VARIABLE SPEED DRIVE ORGANIC FLUID TURBINE (ALRC)	**CURVE TITLE** SYSTEM EFFICIENCY AT 31,200 RPM
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 31



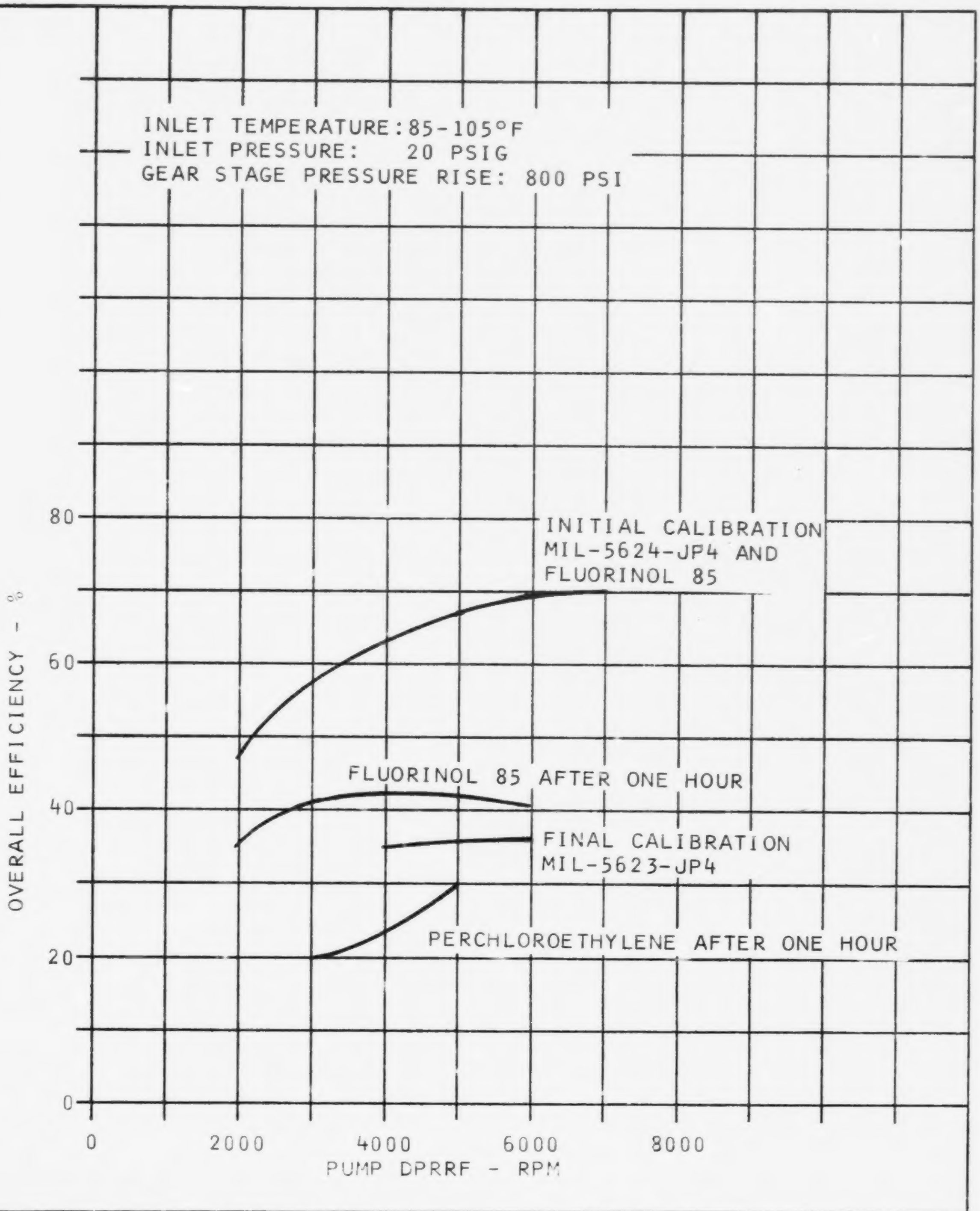
ENG. REPORT NO. APTD-1357	VARIABLE DISPLACEMENT VANE PUMP ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE PUMP EFFICIENCY AT 16,800 RPM
DATE 12-5-72		
PREPARED BY RMR		FIGURE NO. 32



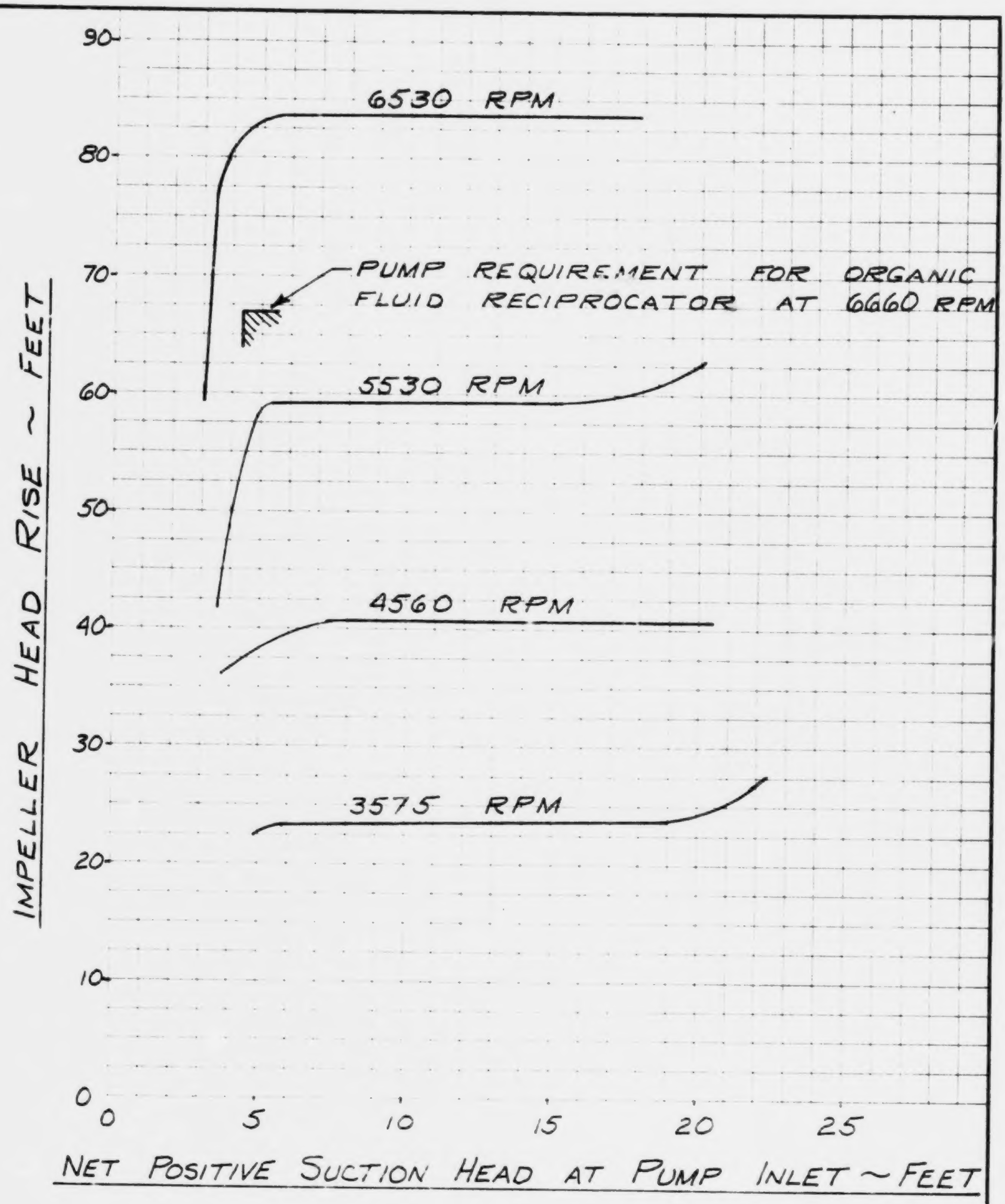
ENG. REPORT NO. APTD-1357	VARIABLE DISPLACEMENT VANE PUMP ORGANIC FLUID TURBINE (ALRC)	CURVE TITLE PUMP EFFICIENCY AT 31,200 RPM
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 33



ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** PUMPING GEAR SCUFFING VS MACHINE LOAD - LBS
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 34



ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** MFP-150-2 PUMP EFFICIENCY VS PUMP SPEED
DATE 12-5-72		
PREPARED BY RMR		**FIGURE NO.** 35



ENG. REPORT NO. APTD-1357

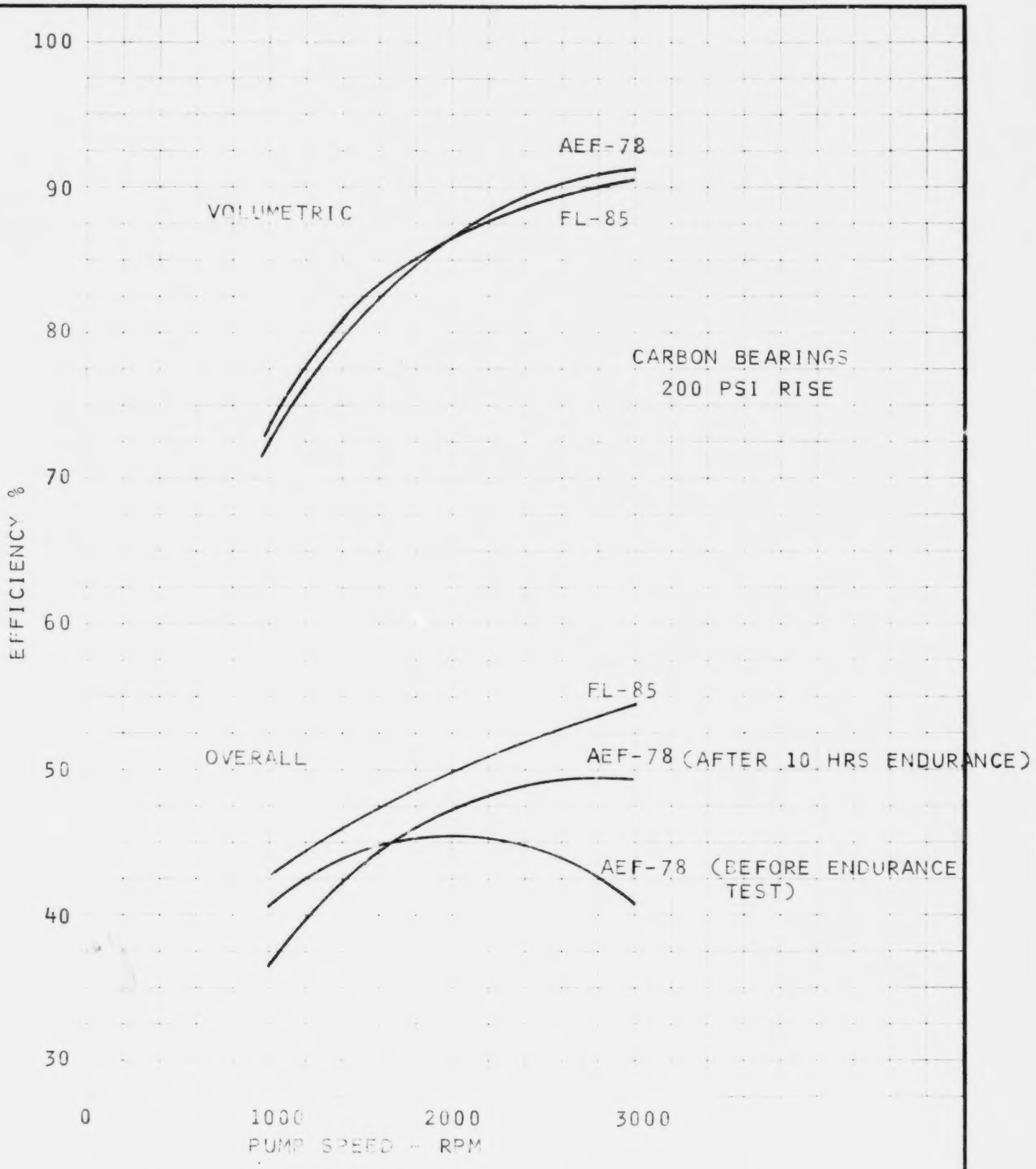
DATE 12-5-72

PREPARED BY RMR

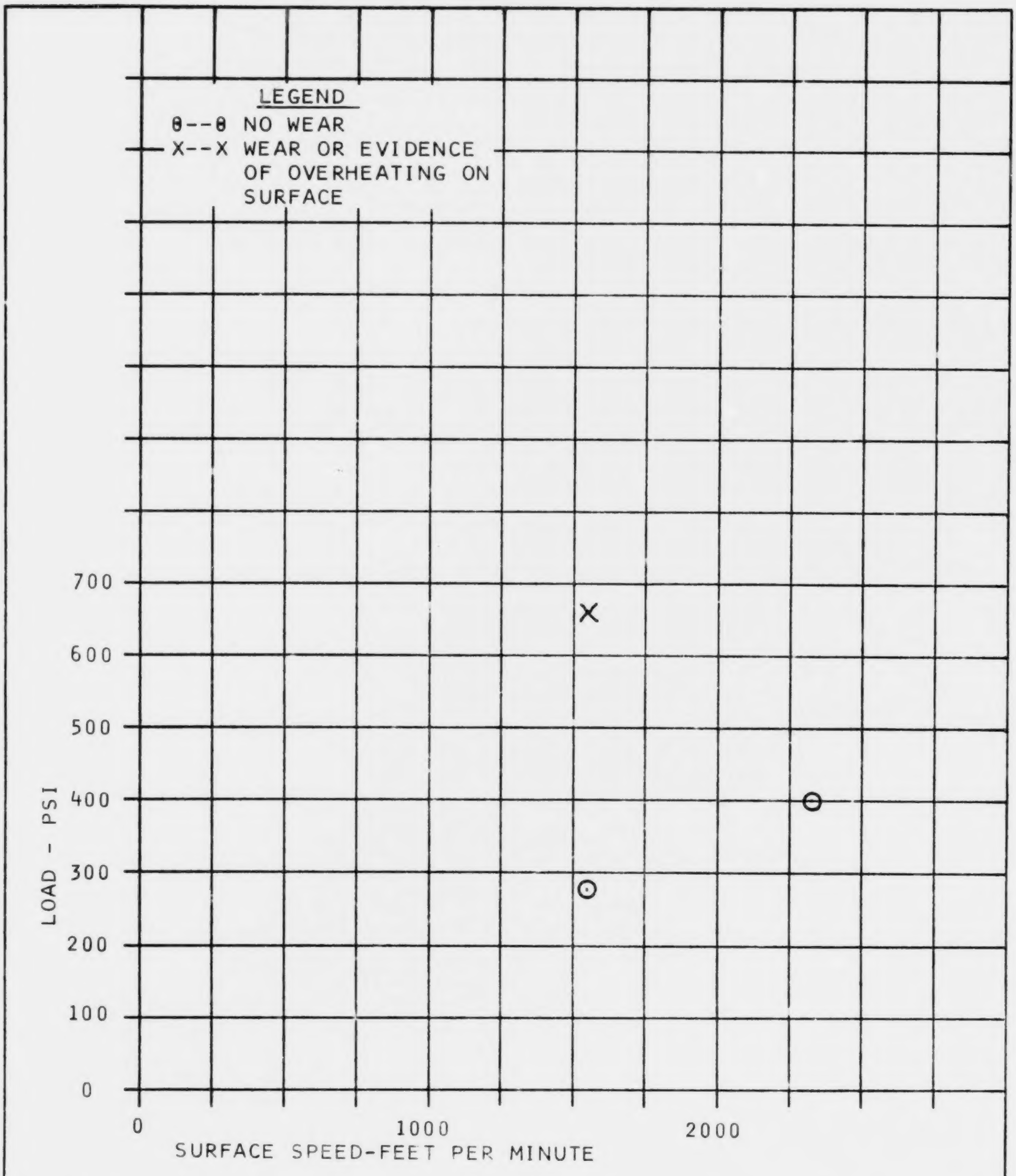
REPORT TITLE MODEL MFP-150 IMPELLER PERFORMANCE

CURVE TITLE IMPELLER HEAD RISE -VS- NPSH

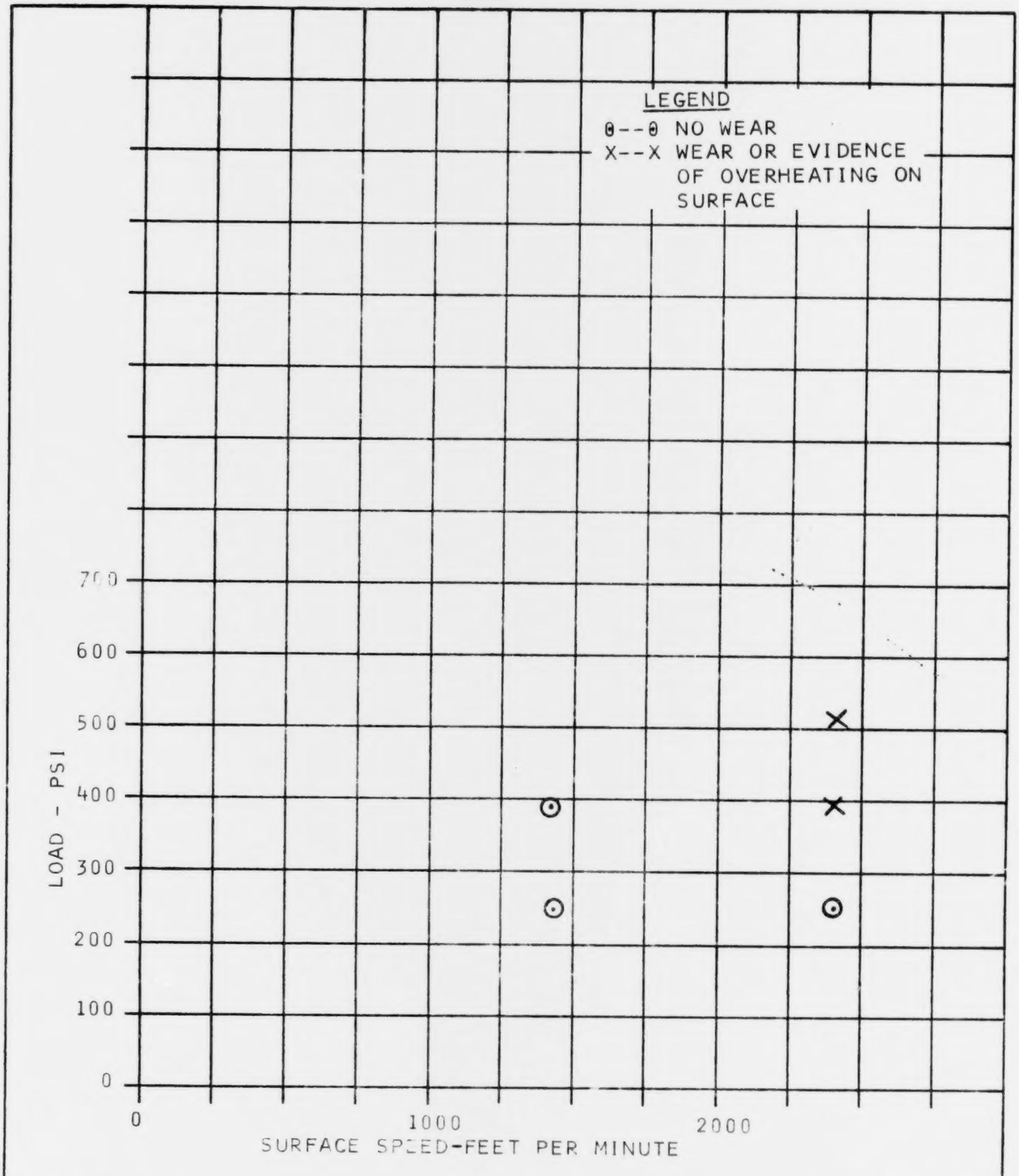
FIGURE NO. 36



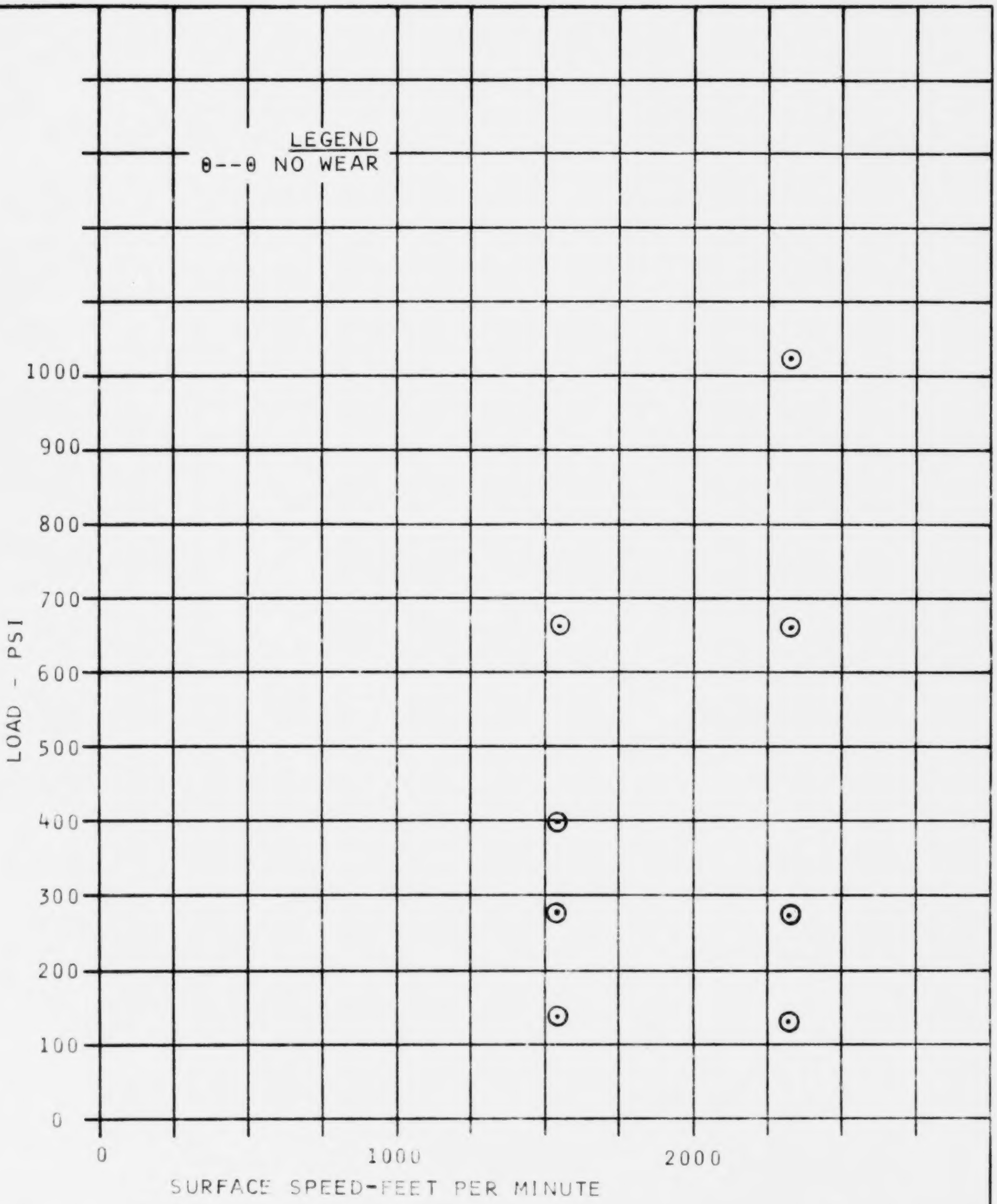
ENG. REPORT NO. APTD-1357	REPORT TITLE	CURVE TITLE MFP-150-2 PUMP EFFICIENCY
DATE 12-5-72		
PREPARED BY		FIGURE NO. 37



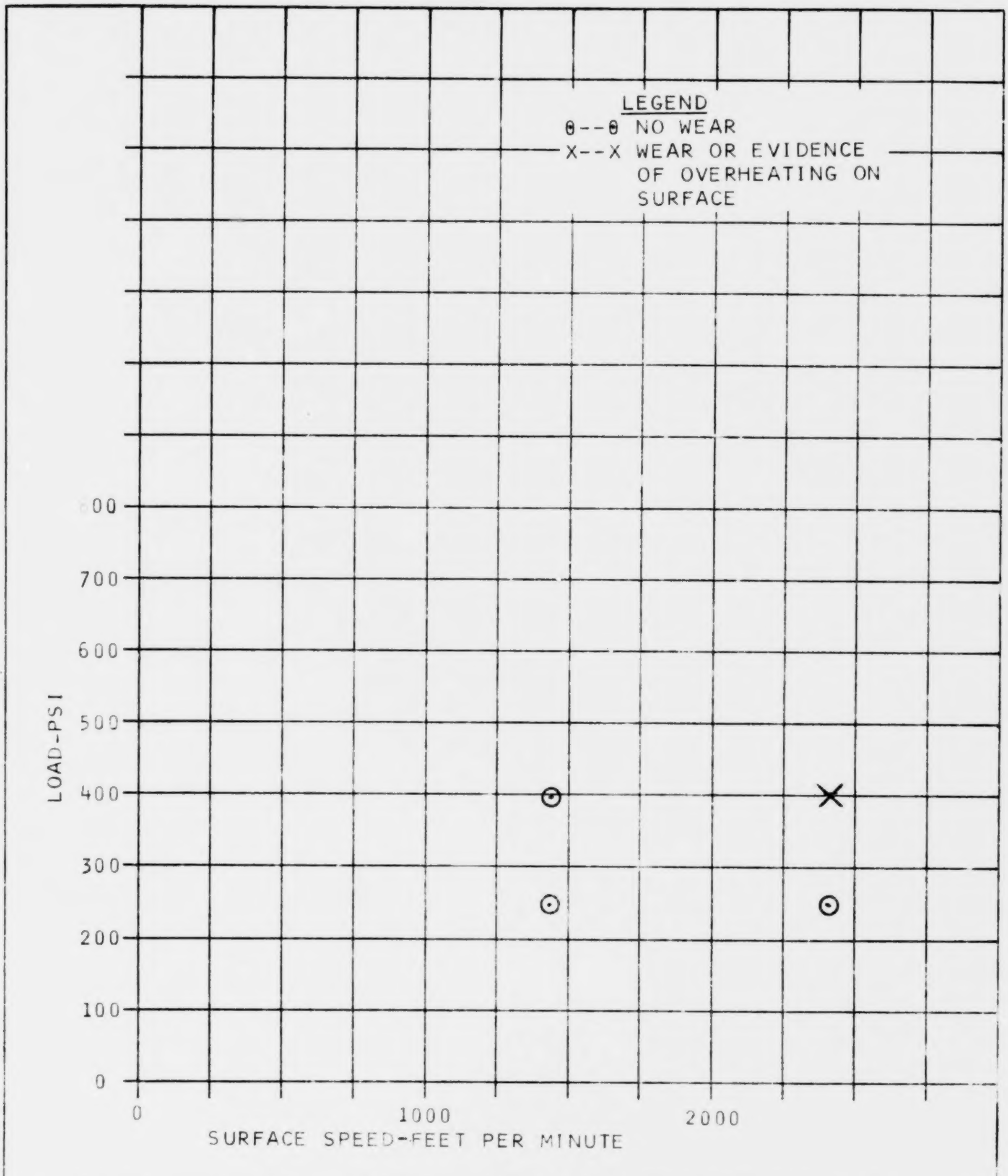
ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** ROTATIONAL WEAR TEST: NITRIDED NITRALLOY VS RESIN IMPREGNATED CARBON GRAPHITE IN FLUORINOL 85
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** 38



ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** ROTATIONAL WEAR TEST:-440C STAINLESS VS RESIN IMPREGNATED CARBON GRAPHITE IN FLUORINOL 85
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** 39



ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** ROTATIONAL WEAR TEST: NITRIDED NITRALLOY VS LEADED BRONZE IN AEF 78
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** 40



ENG. REPORT NO. APTD-1357

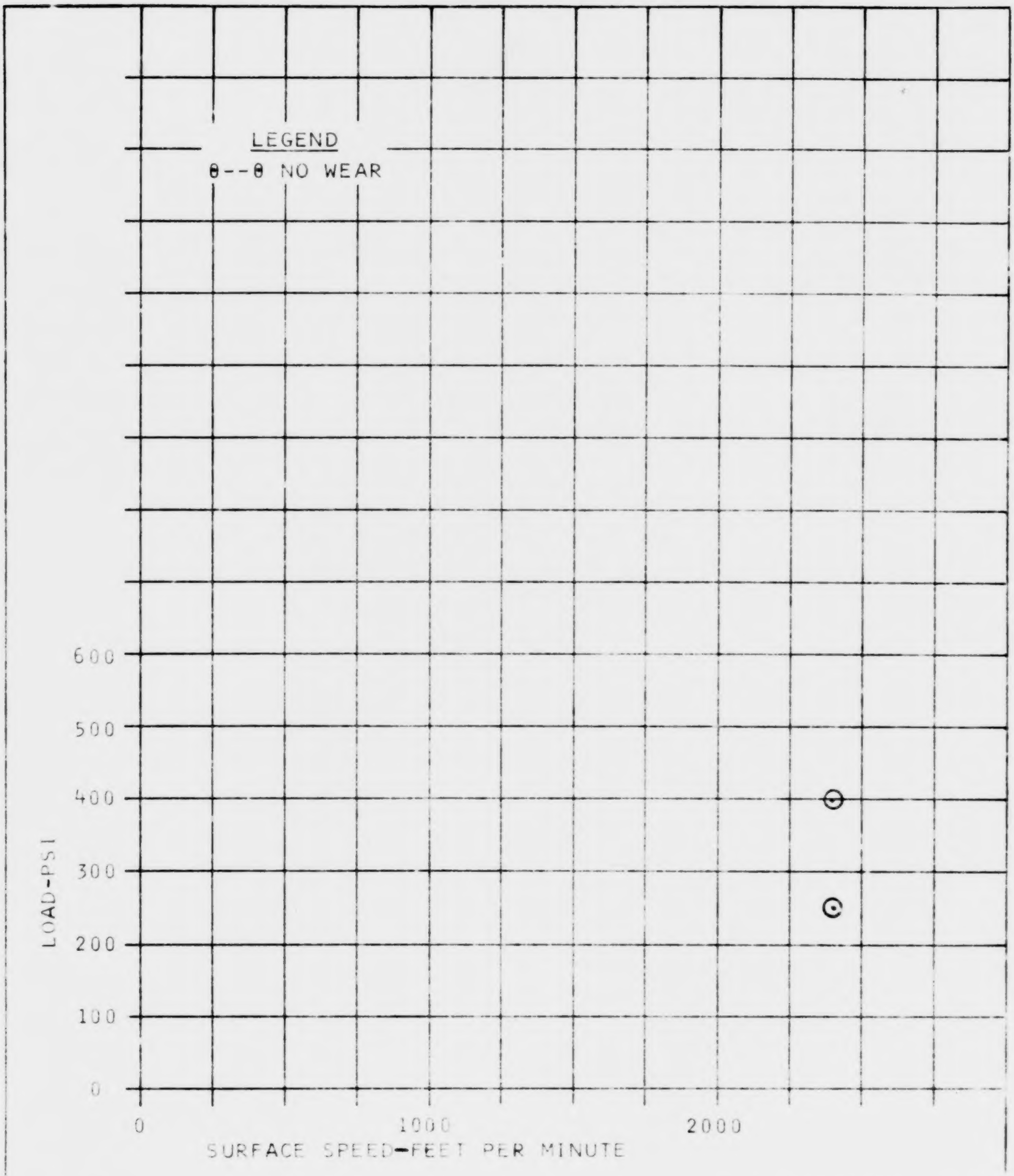
DATE 12-5-72

PREPARED BY

REPORT TITLE

CURVE TITLE ROTATIONAL WEAR TEST: RESIN IMPREGNATED CARBON GRAPHITE VS 440C STAINLESS STEEL IN AEF 78

FIGURE NO. 41



ENG. REPORT NO. APTD-1357

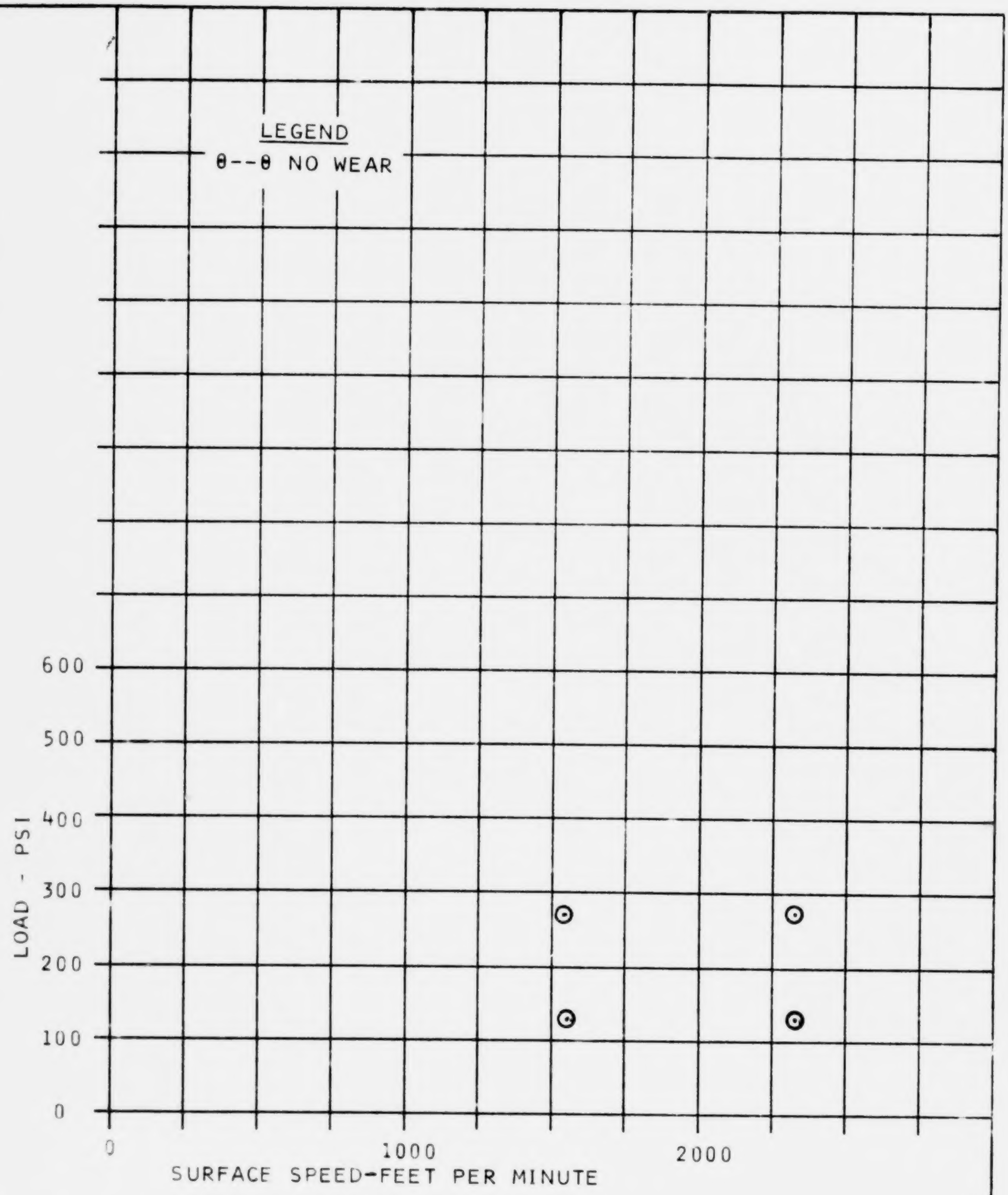
DATE 12-5-72

PREPARED BY

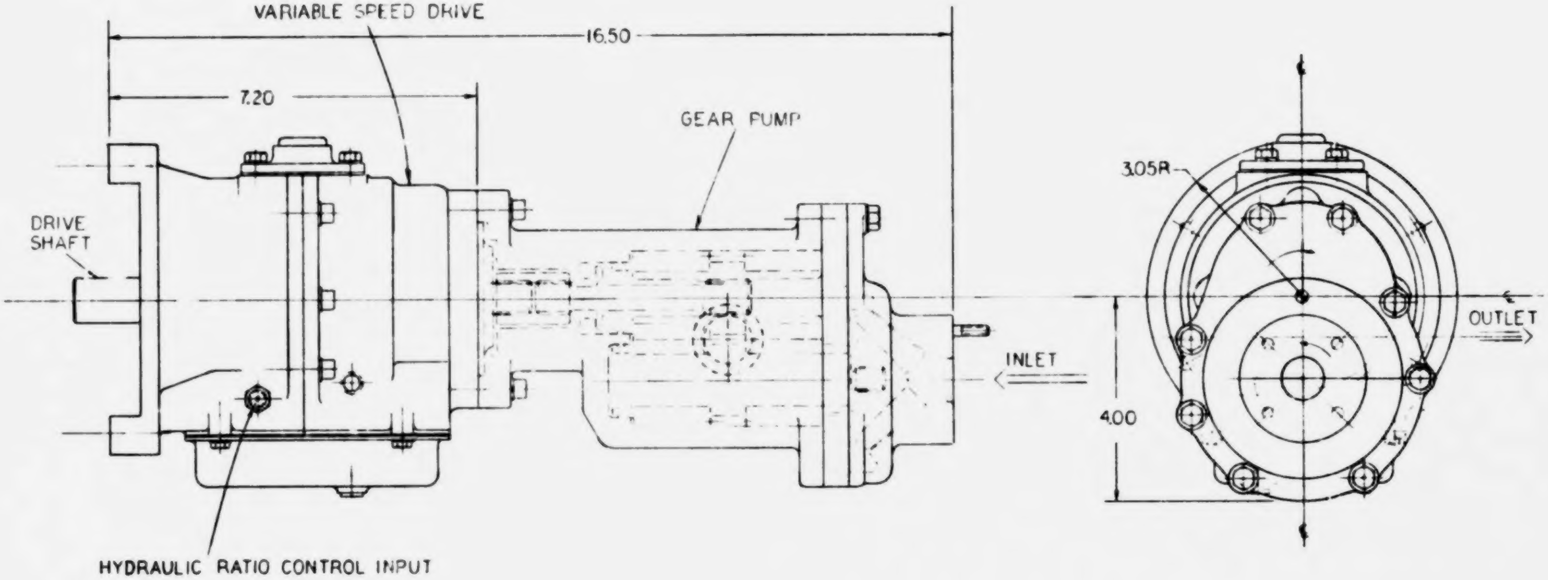
REPORT TITLE

CURVE TITLE ROTATIONAL WEAR TEST: RESIN IMPREGNATED CARBON GRAPHITE VS 440C STAINLESS STEEL IN DISTILLED WATER

FIGURE NO. 42

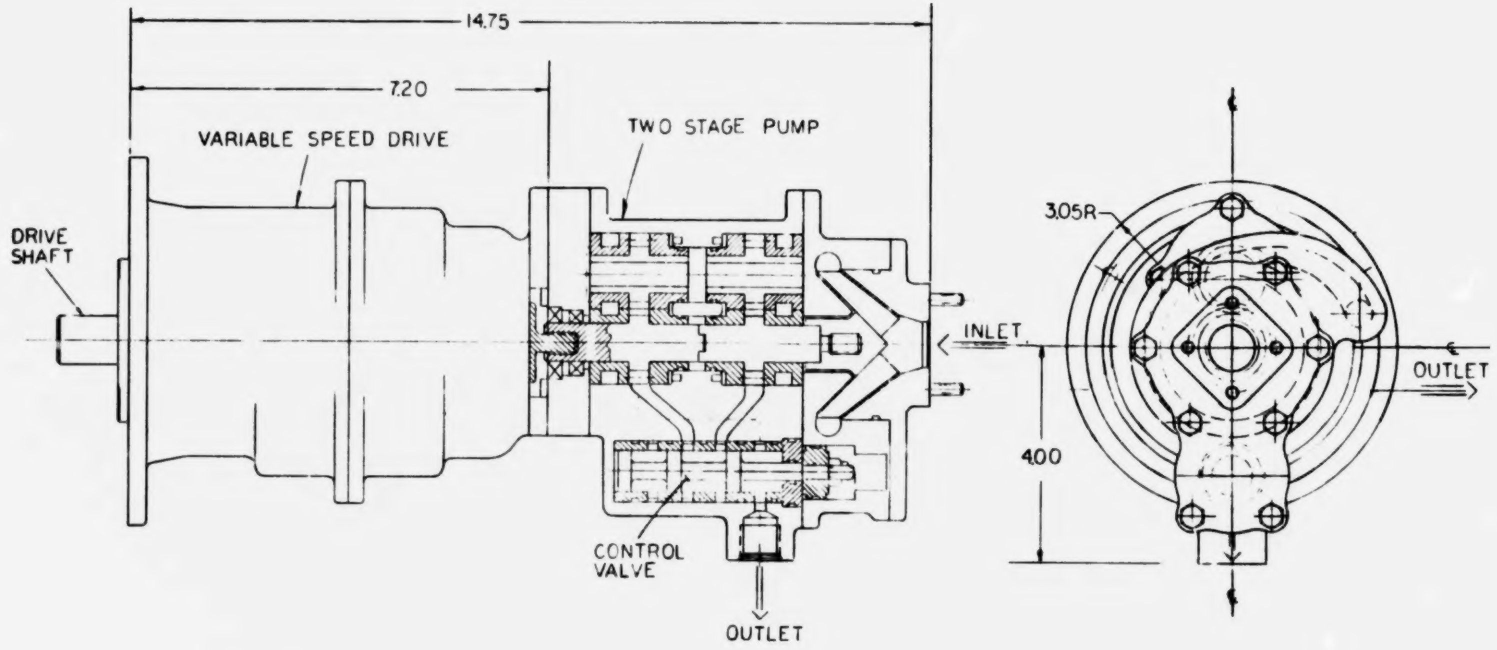


ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** RATIONAL WEAR TEST: NITRIDED NITRALLOY VS LEADED BRONZE IN PERCHLOROETHYLENE
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** 43



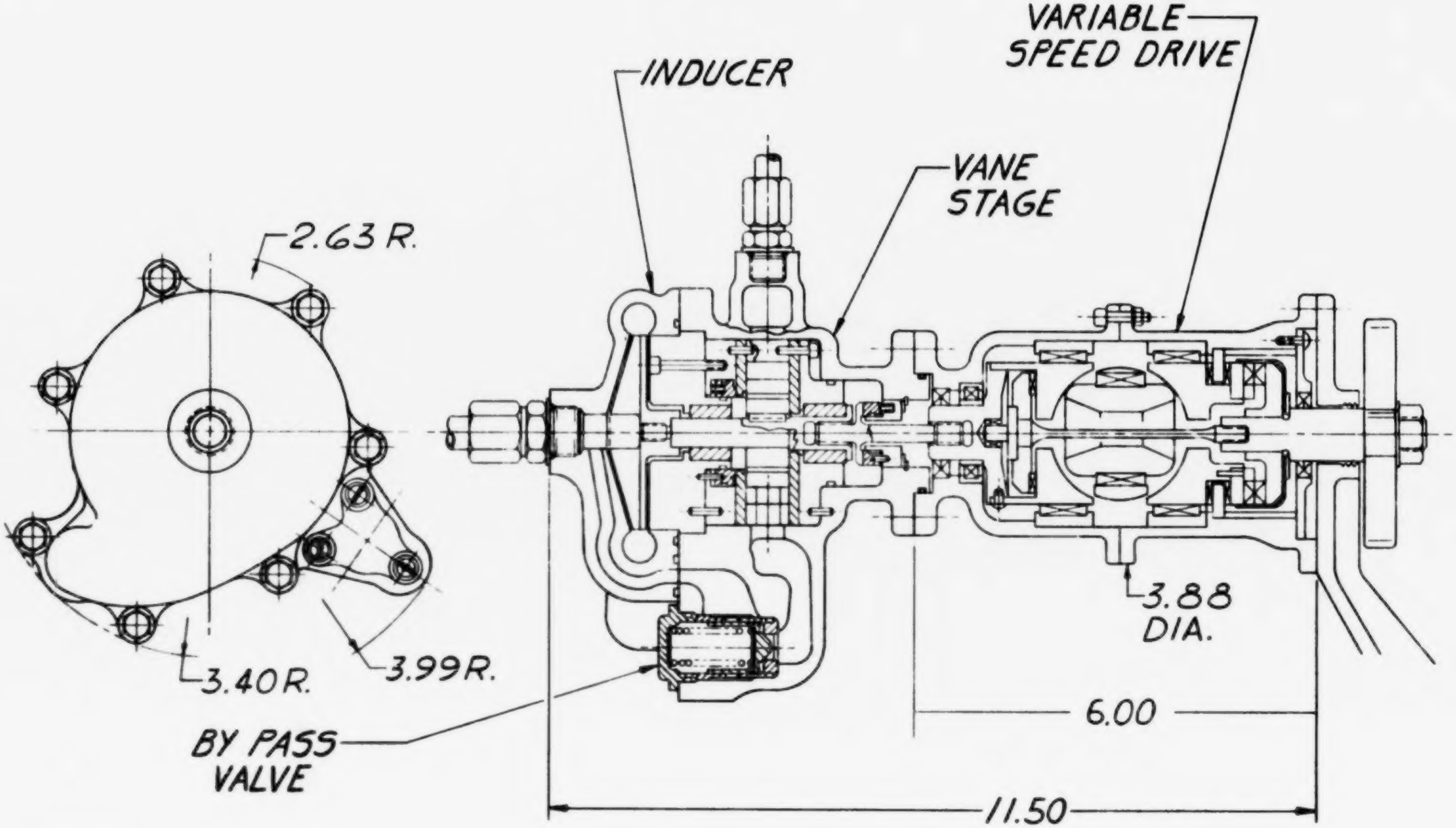
GEAR PUMP ON
VARIABLE SPEED
DRIVE
TECO

Figure 44



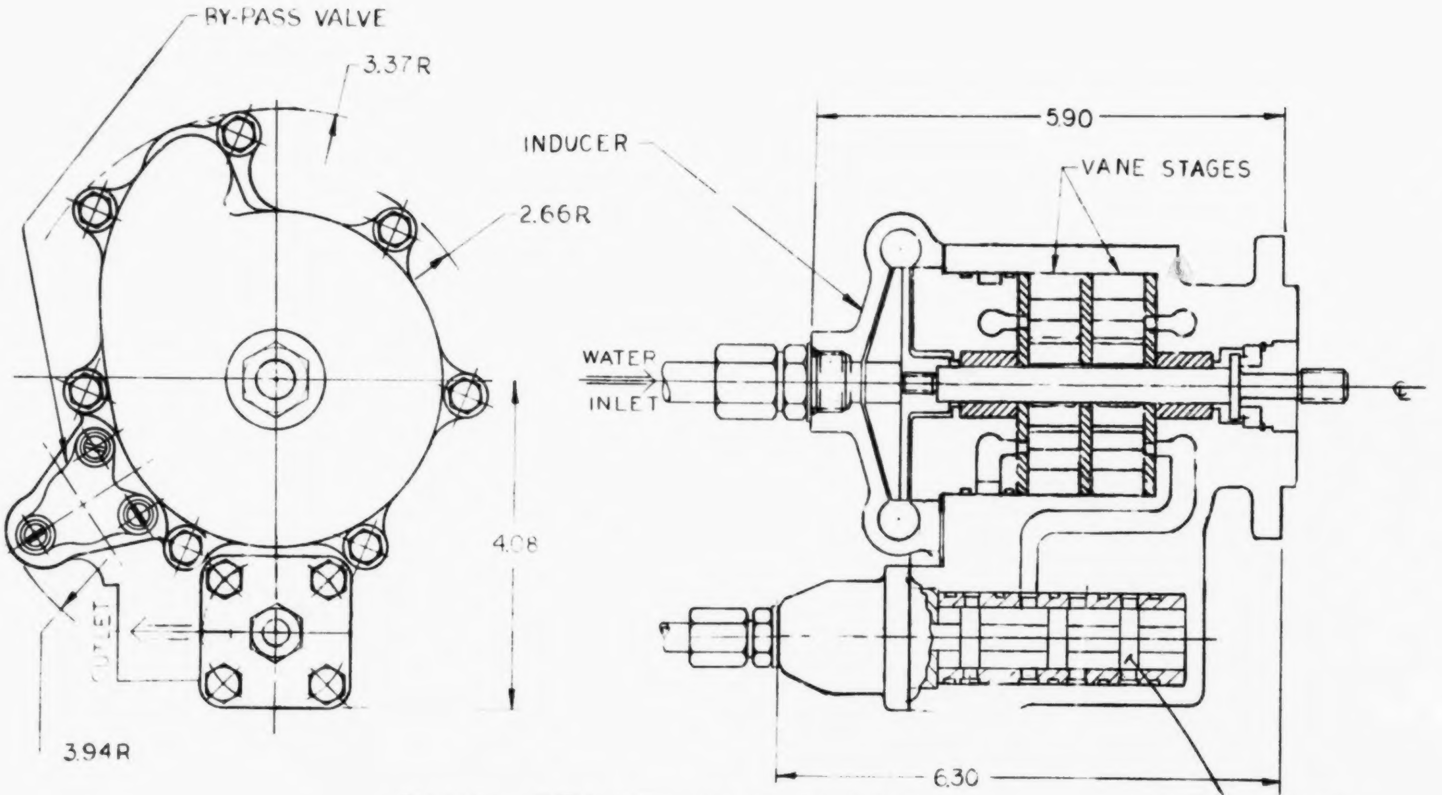
TWO STAGE GEAR
PUMP ON VARIABLE
SPEED DRIVE
TECO

Figure 45



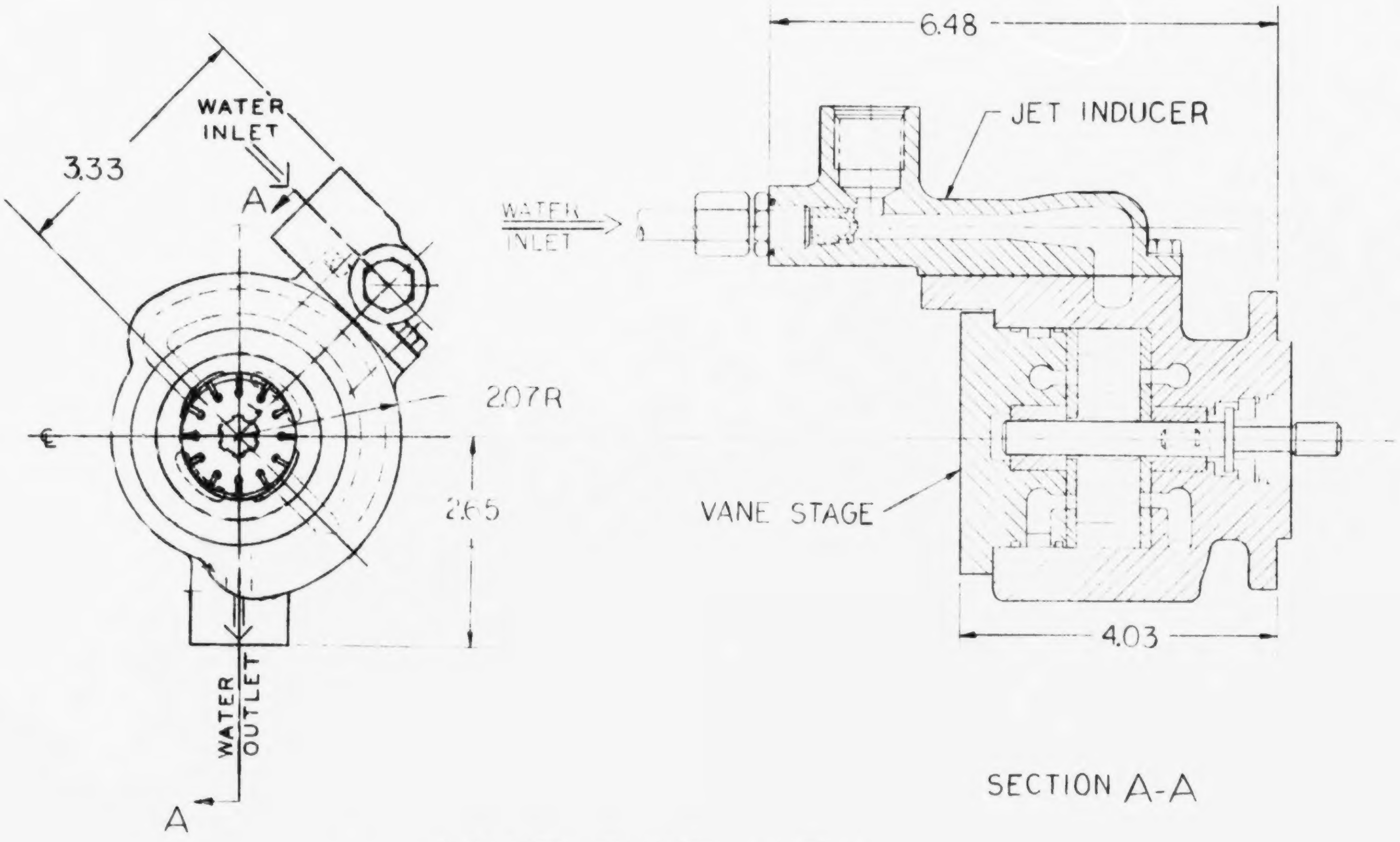
VANE PUMP ON VARIABLE SPEED DRIVE
SES

Figure 46



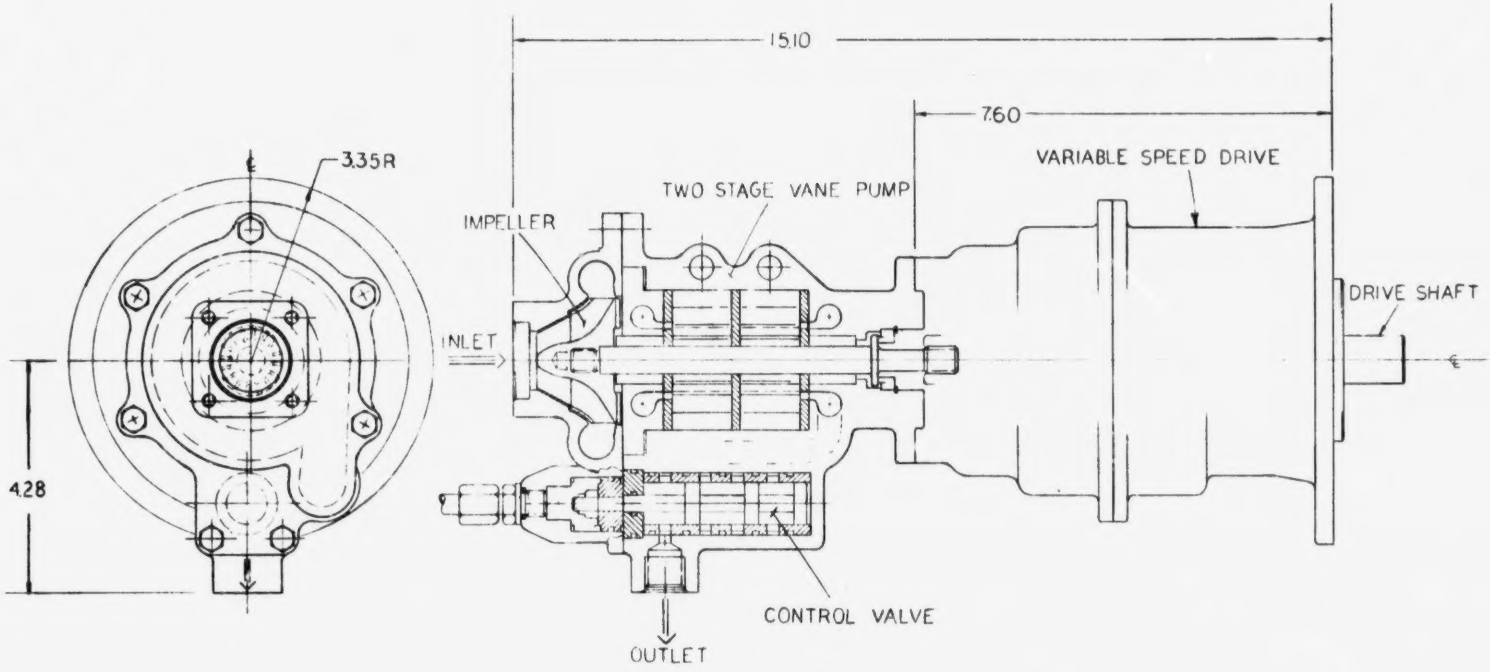
TWO STAGE VANE PUMP
SES

Figure 47



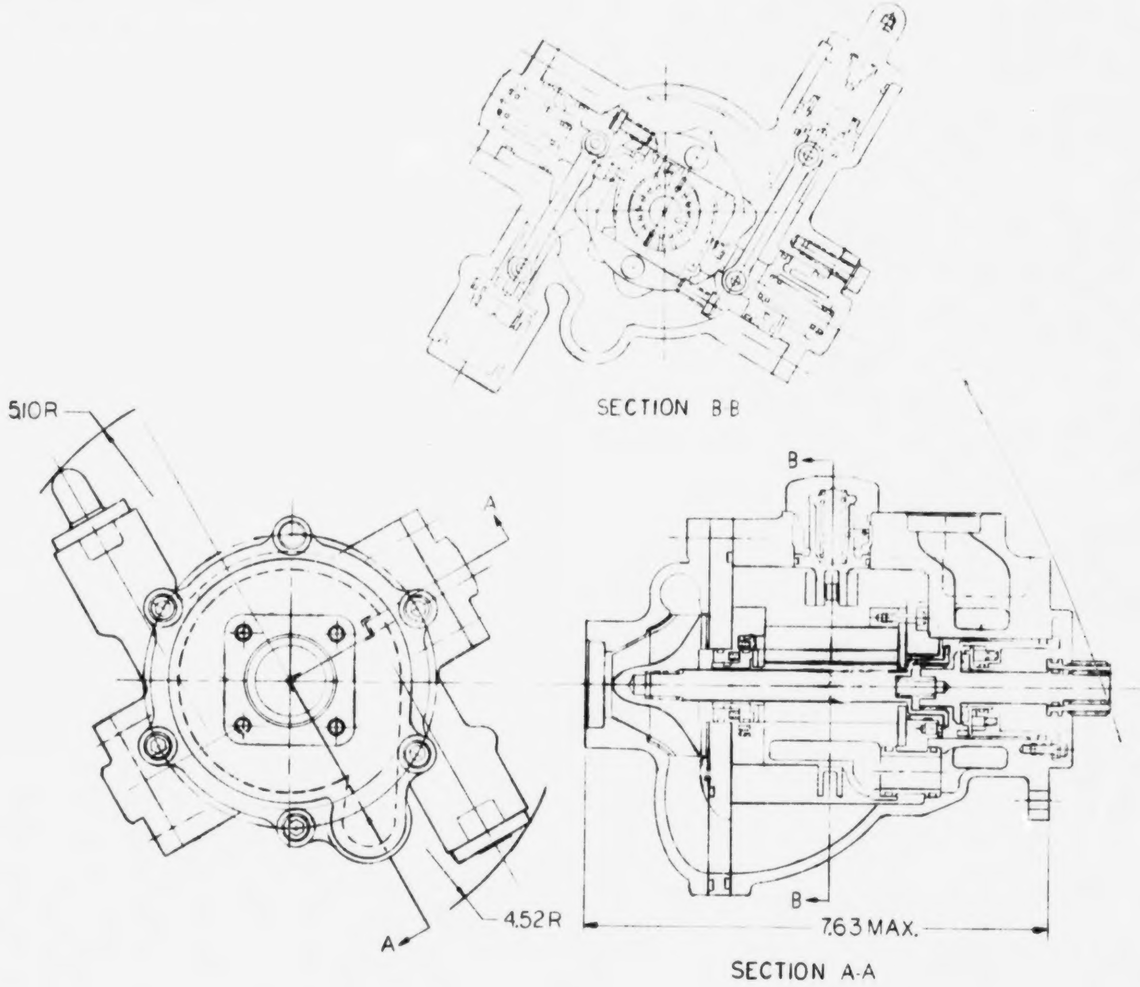
VANE PUMP WITH
INDUCER BYPASS
SES

Figure 48



TWO STAGE VANE
PUMP ON VARIABLE
SPEED DRIVE
ALRC

Figure 49



VARIABLE DISPLACEMENT
VANE PUMP
ALRC

Figure 50

ENG. REPORT NO. APTD-1357 DATE December 5, 1972

APPENDIX A

September 15, 1972

Operation of a
Differential/Variable Speed Drive Combination

130015
REV. 1/65

ENG. REPORT NO. APTD-1357 DATE December 5, 1972

ANALYSIS OF VARIABLE SPEED DRIVE WITH DIFFERENTIAL GEAR TRAIN

The speed range of a variable speed drive can be extended by employing subtracting differential gears (opposite rotation) in the drive train.

To achieve the speed range multiplication desired for the Rankine cycle feed pump systems being studied under EPA Contract No. 68-01-0430, the arrangement shown in Figure A-1 was selected. The variator output speed is multiplied by 4 and the rotation reversed before entering the differential. Drive pad power is fed directly into both the variator and the other differential input.

The effect of employing these parameters is illustrated in the curve of Figure A-2 which has the TECO system (reciprocating expander, organic fluid) requirements imposed upon it. A drive turndown of 12.4:1 can be obtained through a variator turndown of 5:1.

However, the speed range advantage is achieved at the expense of increased power loss and oversizing of the variator.

The analysis contained in the ensuing pages develops this thesis. To better follow the analysis, refer to the schematic of Figure A-1. A bevel gear differential is shown for simplicity.

SYMBOLS

F = Force
T = Torque
r = Radius
η = Efficiency
HP = Horsepower
N = Speed
R = Speed Ratio

Subscripts refer to elements in system.

$$T_A = T_B = \frac{T_{PUMP}}{2} = T_{SP_2} = T_D$$

$$T_{SP_1} = \frac{T_{PUMP}}{2} \left(\eta_{SP + A + C} \right)$$

$$HP_{SP_1} = \frac{T_{SP_1} \; N_{SP_1}}{63000} = \frac{T_{PUMP} \; N_{SP_1} \left(\eta_{SP + A + C} \right)}{2(63000)}$$

$$F_D r_D = T_D$$

$$F_E r_E = T_E$$

$$F_E = F_D$$

$$\frac{r_E}{r_D} = R_G$$

$$T_E = F_D r_E = F_D r_D R_G = \frac{T_D}{r_D} r_D R_G = T_D R_G$$

Consider "A" leg of circuit removed. The "B" leg will, therefore, only transmit power to drive the pump.

$$T_E = \frac{R_G \, T_{PUMP}}{2 \, \eta_{B + C + D + E}} = T_{V_2'}$$

$$HP_{V_2'} = \frac{T_{V_2'} \, N_{V_2}}{63000} = \frac{R_G \, T_{PUMP} \, N_{V_2}}{2(63000)(\eta_{B + C + D + E})}$$

$N_{V_2} = R_V N_{V_1}$

R_V = Variator Speed Ratio

$\eta_V = f(R_V)$

$$HP_{V_1}' = \frac{HP_{V_2}'}{\eta_V}$$

$$HP_{V_1}' = \frac{R_G T_{PUMP} N_{V_1} R_V}{2(63000)\, \eta_V\, (\eta_{B+C+D+E})}$$

$N_{V_1} = N_{SP_1} = N_{DRIVE\ PAD}$

Since the power in the "A" leg is recirculating, it must also be operated on by the efficiency in the "B" leg.

∴ Total HP in variable speed drive (VSD)

$$= HP_{V_1} + \frac{HP_{SP_1}}{\eta_V\, (_{B+C+D+E})}$$

$$HP_{VSD} = \frac{T_{PUMP} N_{DRIVE\ PAD}}{2(63000)} \left[\frac{\eta_{SP+A+C} + R_G R_V}{\eta_V\, (\eta_{B+C+D+E})} \right]$$

MFP-150 ON VARIABLE SPEED TRANSMISSION WITH DIFFERENTIAL

Condition (TECO)	N_{DP}	N_P	R	Q	ΔP	η_P	HP_P	T_P	R_G	R_V	η_V (TPI)	η_A	η_B	HP_{VSD}
1	900	1065	1.182	1.3	850	24	2.685	159	4	.841 (1/1.19)	92.2	98	97	5.51
2	900	3225	3.59	7.3	850	71.5	5.06	98.8	4	2.045	89.6	98	97	7.44
3	1800	5852	3.25	14.6	850	79	9.15	98.4	4	1.875	89.9	98	97	13.7
4	2400	6716	2.8	17	850	78.5	10.72	100.7	4	1.65	90.3	98	97	16.6
5	6000	6716	1.12	17	850	78.5	10.72	100.7	4	.81 (1/1.233)	86	98	97	24.3
6	6000	1750	.292	3.2	850	52.5	3.02	108.7	4	.396 (1/2.52)	85.3	98	97	16.03

RECIPROCATING EXPANDER - ORGANIC FLUID (TECO)

DRIVE TRAIN EFFICIENCY

$$\eta_{DRIVE} = \frac{HP_{OUT}}{HP_{IN}} = \frac{HP_{P}}{HP_{VSD} - HP_{RECIRCULATING}} = \frac{HP_{P}}{HP_{VSD} - HP_{SP,}}$$

$$HP_{SP,} = \frac{T_{P}\ N_{DP}\ \eta_{SP} + C + A}{2(63000)} = \frac{T_{P}\ N_{DP}}{128,571}$$

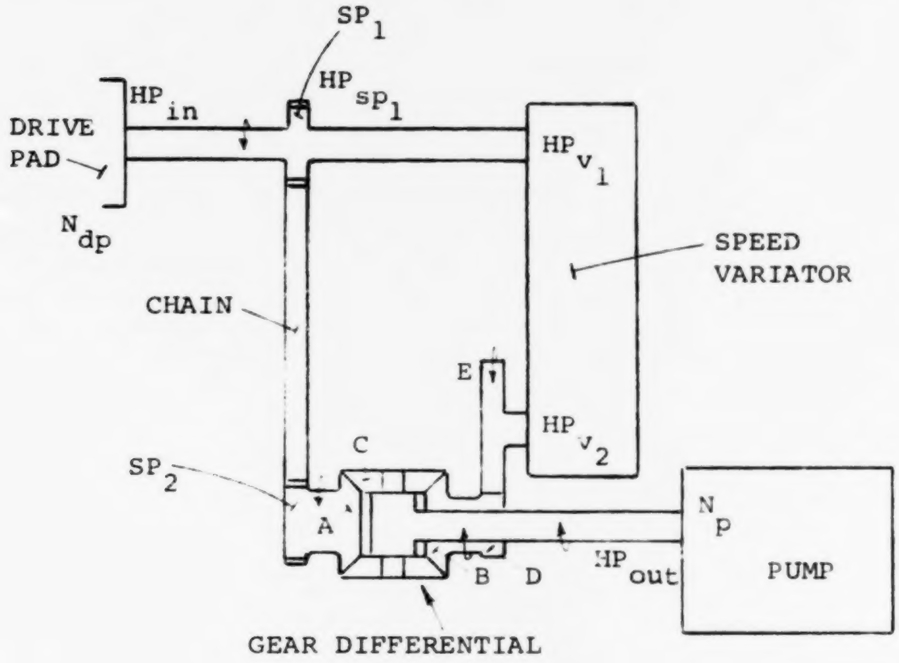
Condition	T_P	N_{DP}	$HP_{SP,}$	HP_P	HP_{VSD}	$HP_{VSD}-HP_{SP,}$	η_{DR} %
1	159	900	1.09	2.685	5.51	4.42	60.8
2	98.8	900	.692	5.06	7.44	6.75	75.1
3	98.4	1800	1.377	9.15	13.7	12.32	74.3
4	100.7	2400	1.88	10.72	16.6	14.72	72.9
5	100.7	6000	4.7	10.72	24.3	19.6	54.7
6	108.7	6000	5.07	3.02	16.03	10.96	27.55

SYSTEM EFFICIENCY

Condition	η_{DR} %	η_{PUMP} %	η_{SYSTEM} %
1	60.8	24	14.6
2	75.1	71.5	53.6
3	74.3	79	58.6
4	72.9	78.5	57.1
5	54.7	78.5	42.9
6	27.55	52.5	14.44

SYSTEM EFFICIENCY COMPARISON WITH AND WITHOUT THE DIFFERENTIAL

Condition	η System with Differential	η System without Differential
1	14.6	22.1
2	53.6	64.0
3	58.6	71.0
4	57.1	71.0
5	42.9	67.5
6	14.4	44.8



VARIABLE SPEED DRIVE WITH DIFFERENTIAL GEAR TRAIN

FIGURE A-1

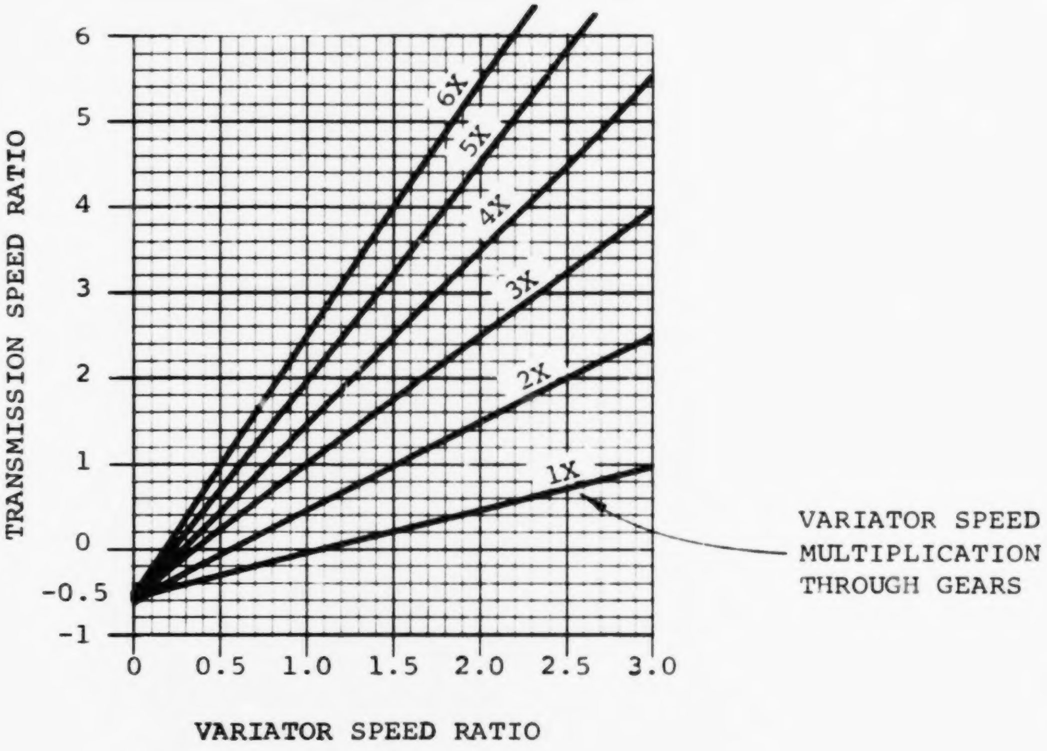


FIGURE A-2

APPENDIX B

TECO Seal Compatibility Tests

THERMO ELECTRON
CORPORATION

85 FIRST AVENUE
WALTHAM MASS 02154
TEL 617 890-8700
CABLE TEECORP

September 13, 1972

L.A. Difford
Colt Industries, Chandler Evans
Control Systems Division
Charter Oak Boulevard
West Hartford, Connecticut 06101

Dear Bud:

The compatability tests of your samples with Fluorinol-85 have been completed and the results are shown in the enclosed memorandum from our chemist to Dean Morgan.

Test conditions are listed and the results as well as conclusions and recommendations are given. Also included is a list of materials recommended for use by TECO.

Yours truly,

Luco R. DiNanno
Development Engineer

mg
Enclosures

cc: K. Barber

MEMORANDUM

DATE: September 5, 1972

TO: Dean Morgan

FROM: Alexandra Zakak

SUBJECT: Compatibility Test of Some Rubber Products

Summary

Samples of preformed packings* of both fluorosilicone and silicone compounds were tested in F-85 and a lubricating oil (v. ratio 10/1) in the presence of air at 250°F. After a testing period of 17 days (continuous heating) both fluorosilicones (LS53 and LS63) were much softer and were lightly attacked at the bottom in F-85 immersed portion. The silicon compound no. S455-7 showed to be more resistant in F-85. A light softening occurred in the portion of the ring immersed in liquid F-85. Conclusions and recommendations are also given.

Results

Test conditions and results of visual observation of tested samples are summarized in Table I. Each sample was sealed in air in a pyrex test capsule. Both fluorosilicones samples LS53 and LS63 were attacked in the bottom immersed part and were much softer. The silicone compound no. S455-7 was very lightly softer, but gained some weight due to swelling, about 11%, with a light change in dimensions. The liquid F-85 turned to a very light or light brown color.

Conclusions

1. Fluorosilicone rubber, Dow Corning LS53 was attacked at the bottom portion of the sample, immersed in F-85. The material became much softer.

2. Fluorosilicone rubber, Dow Corning LS63 was very lightly attacked at the bottom portion of the sample, immersed in F-85. The material became much softer.

3. Silicone rubber, Parker no. S455-7 seemed not to be attacked by F-85 liquid or vapor, percent swelling was approximately equal to 11%. No major change in dimensions occurred.

---

*See Appendix for information on manufacturers.

TABLE I

TEST CONDITIONS

Test Conditions: Solution: F-85 and lubricating oil (v. ratio 10/1)
Temperature: 250°F
Duration: 17 days

Specimen No.	Material	Atmosphere in seal	Visual Observation
779*	Dow Corning LS53, fluorosilicone rubber (~ 2" x 0.07" O. D.)	air	the rubber was much softer, lightly attacked at bottom in liquid immersed portion, brownish material was sticking on glass and liquid F-85 turned to very light brown
780*	Dow Corning LS63, fluorosilicone rubber (1.5" x 0.1" O. D.)	air	the rubber was much softer, lightly attacked in liquid immersed portion. F-85 turned very light brown.
781**	Parker No. S455-7 silicon rubber (2" x 0.1" O. D.)	air	the rubber was lightly softer, no apparent change, F-85 turned light brown.

*After test, dimensions changed
**Weight change after test + 0.0425 gm
% sweiling ~ 11%

Recommendations

1. As a result of the static capsule test, it is recommended to use silicone rubber Parker no. S455-7 in the presence of F-85 at 250°F.
2. Compatibility tests must be run previous to any use of material if higher temperatures are expected.

COMPATIBILITY TESTS OF SOME ELASTOMERIC AND NON-METALLIC MATERIALS

Recommendations

1. Teflon (TFE) is also recommended to use at temperatures not exceeding 350°F in the presence of F-85 solution and air.

2. Teflon fiber (DuPont De NeMours Co., Textile Fibers Department), hard or soft packing may be used up to 250°F in the presence of F-85 solution in air (respective % swelling ~ 35% and 13%). It is believed that Teflon fibers can stand temperatures up to 350°F in the presence of F-85 and air. More tests are needed.

3. Al-Si-Mag 840 ceramic (American Lava Co.) may be used up to 350°F in the presence of F-85 and oil.

4. Glass and mineral filled phenolic No. FM4005 (Fiberite Corp.) may be used in the system up to 200°F in the presence of F-85 and oil.

5. Silicone rubber (o-ring) Parker No. S455-7 is recommended to be used in the system at temperatures not exceeding 250°F in the presence of F-85 solution and air.

APPENDIX C

Gear Pump Design

Pump Size

The displacement and proportions of the pump must be selected to offer the best efficiency over the largest range of speed and flow operation with pressure rise constant.

Displacement

The peak overall efficiency of a positive displacement pump varies with size. The cause of this variation is the effect of manufacturing tolerances and operating clearances, which vary little with size, and windage and porting losses associated with high capacity machines. Gear pumps for low viscosity applications follow a trend as shown below.

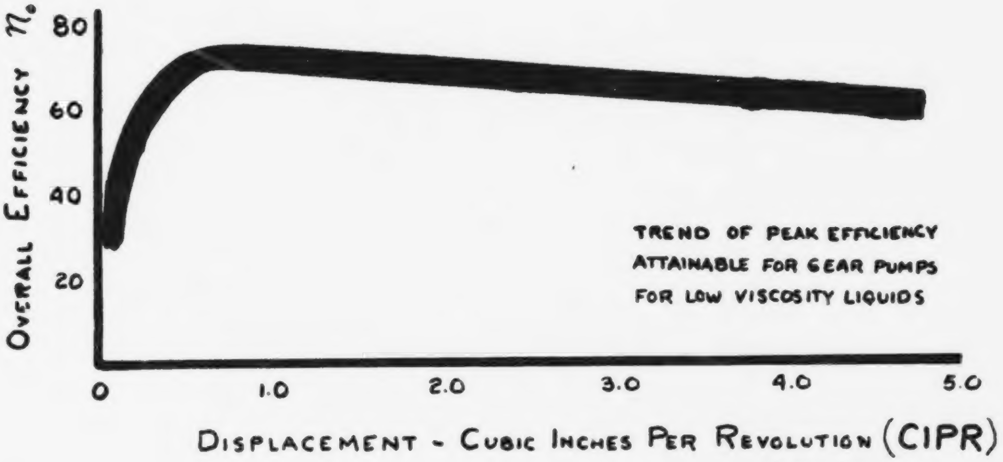


FIGURE C-1

This peak efficiency occurs at a certain pressure rise at each speed, the ratio of which remains fairly constant. The following figure illustrates a typical variation of a gear pump efficiency with the speed normalized pressure parameter.

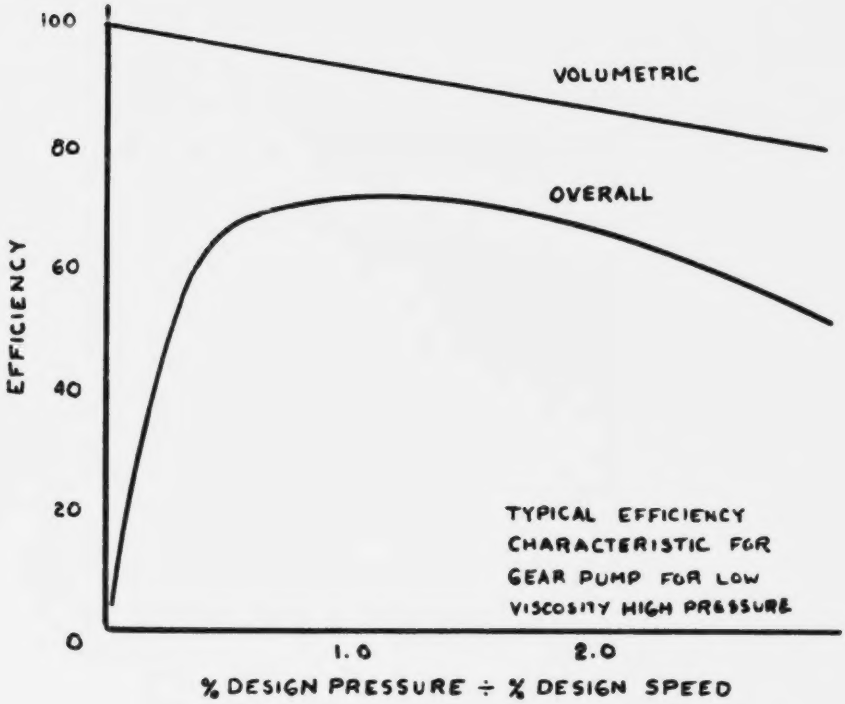


FIGURE C-2

The absolute value of the pressure rise to speed ratio for peak efficiency has been found to vary with displacement as shown in the following figure.

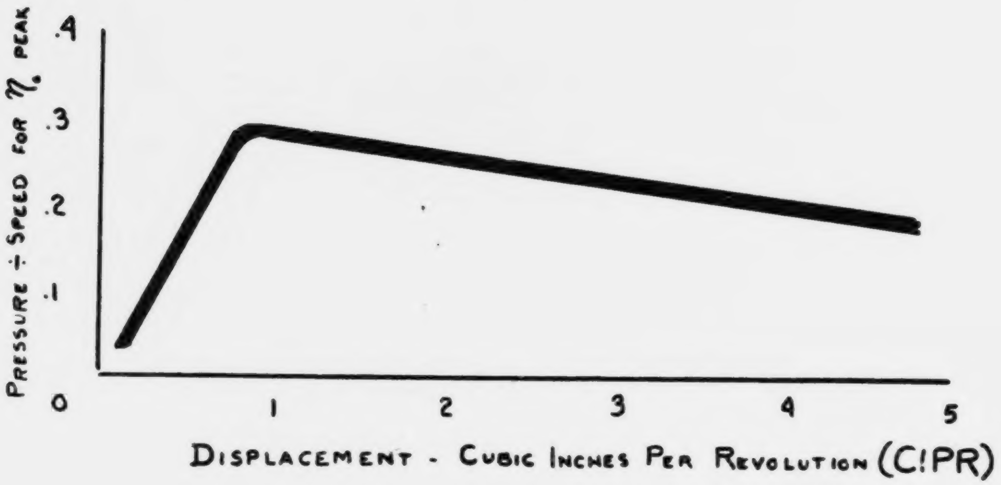


FIGURE C-3

From these considerations, it is apparent that for acceptable overall efficiency, the pump displacement should be between .2 and 2.0 CIPR. The selection is dependent on an iterative analysis which includes:

1) The effect of volumetric efficiency on sizing for maximum flow and for minimum speed.

2) The variation of pump efficiency with speed (reference Figure C-2).

3) The effect of maximum speed on minimum NPSP.

4) The effect of maximum speed on wear.

As a result of this analysis, the following capacity ranges were selected for the steam and organic systems. The limits imposed on these ranges are noted. The actual size selection within this range is made in the following proportioning analysis.

System	CIPR	Limits for Sizing
Steam System (2.5 gpm nominal)	.2 Min.	Smaller units produce low overall efficiency
	.6 Max.	Larger units have poor low speed volumetric efficiency
Organic Systems (25 gpm nominal)	.5 Min.	Smaller (high speed) units require high NPSP
	1.0 Max.	Larger units have low overall efficiency

Proportioning

The desired efficiencies discussed in the Displacement Section can only be achieved if the pump proportions are correct. Those proportions critical to a gear pump design are as follows:

L/D_G = The gear length to diameter ratio which must remain between 1 and .25 for acceptable volumetric efficiency.

L/D_J = The gear journal length to diameter ratio which is only effective up to a limit of 2.

DP = The diametral pitch which produces poor gear action at values below 8 and inefficient packaging at values over 12.

T = The number of gear teeth which must remain between 8 and 20 for the same reasons cited for DP.

D_J/D_R = Ratio of gear journal to root diameter which must be less than .75 for effective sealing.

These proportions must be matched against the required pressure rise and flow rate requirements while still satisfying the following limiting parameters for the materials described in the Materials Section.

UBL = The unit load on the journal < 600 psi

PV = The combined load on the journal $< 1.5 \cdot 10^6$ (psi)(fpm)

PVt = The scoring factor for the gear $< 300{,}000$
(t = length of path of contact of gears)

U = Gear tip speed which dictates the boost pump charging pressure required

An 18-tooth, 10 DP. gear was selected for both the organic cycles. This gear offered the lowest stress levels throughout the gear and journal while requiring a net positive suction pressure which can adequately be achieved by a boost pump. Both systems use the same (.9 CIPR) displacement for the single-element design while the dual element concept reduces the length of this same gear to effect a .6 CIPR.

Pump Performance

Based on data from gear pumps of the same displacement, the following performance is predicted for the organic turbine cycle gear pump.

Efficiency

The overall and volumetric efficiencies of these pumps are predicted in the following figure.

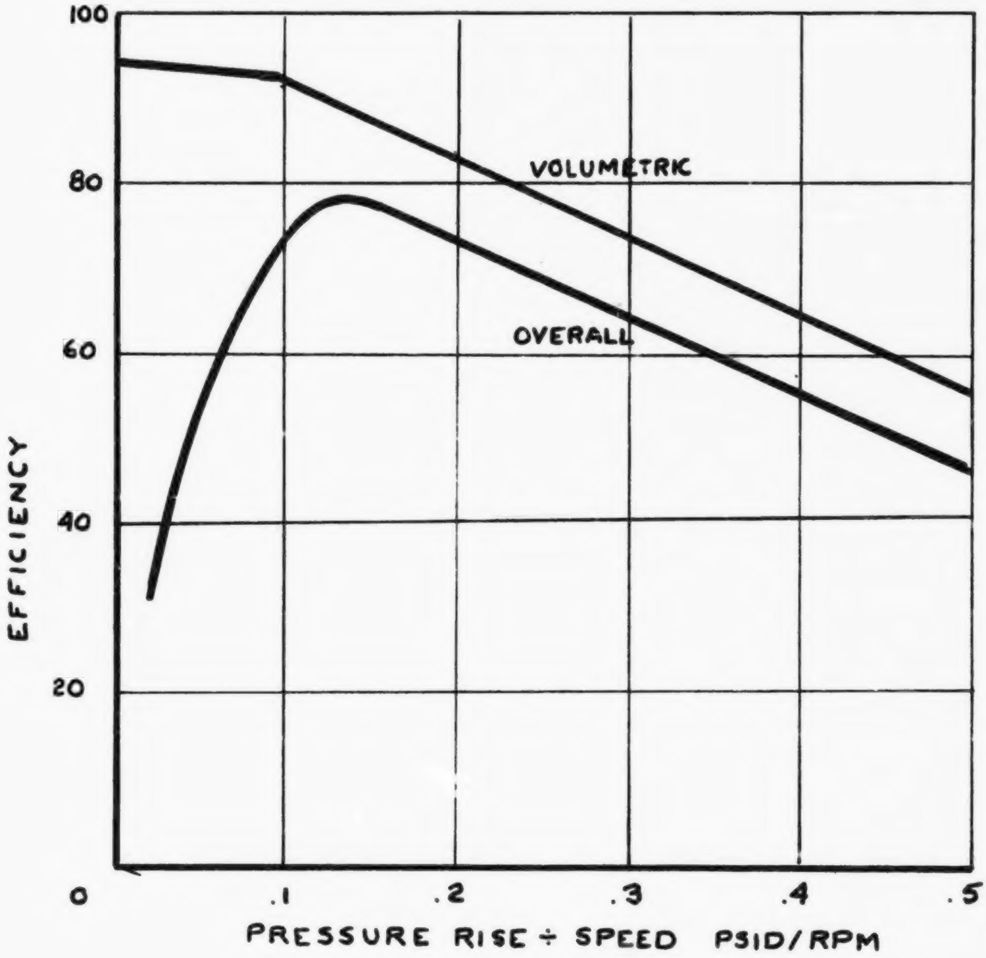


FIGURE C-4

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APPENDIX D

Vane Pump

Pump Sizing

The pump displacement and proportions must be selected to offer acceptable system efficiencies. The analytical approach is similar to that used for the gear pump (Appendix C), however, since there is not as much experience with vane pumps in high pressure applications with low viscosity fluids, predicted performance is based more on a detail loss analysis.

Displacement

Test results for vane pumps in low viscosity service agree with the more extensive experience previously presented in Appendix C for Gear Pumps. This agreement occurs both in the peak efficiency to be achieved vs. displacement (Figure C-1) and on the location of this peak efficiency in relation to the pressure rise vs. speed ratio (Figure C-3). It follows, then, that the displacement range for further proportioning analysis should be the same as that chosen for the Gear Pumps.

Proportioning

The following proportioning ratios have been established for the vane pump for the reasons cited.

S/H = Vane stroke to height ratio selected at a nominal value of .32. Higher values present problems of vane guidance in the slot, and lower values adversely effect the rotor strength.

S/W = Vane stroke to width ratio selected at a nominal value of 1.6 since both higher and lower values produce rotor strength problems.

AR = Ratio of vane track acceleration to the centripetal acceleration selected at .6 since the vane cannot follow the cam at higher values, and the cam loads are undesirably high at lower values.

These ratios are input to a computer program along with the required performance at flow rate and pressure rise at a selected operating speed. Material properties and mechanical sizing are either input or monitored at the output to assure sound mechanical design. These parameters include vane bending and Hertz stress. The parasitic losses within the vane stage are calculated and later summed with the predicted bearing and seal losses to allow prediction of total power consumption and resulting overall efficiency.

The result of an iterative analysis of several computer design studies indicates that the most favorable design for the Steam Engine is that presented on the following computer readout sheet.

```
 VANE STAGE PROPORTIONING ANALYSIS CHG A


INPUT DATA
    INPUT VANE SPEED................  5000.00  RPM
    THEORETICAL DISPLACEMENT.......    0.3995  CU IN/REV
    SLIDING VELOCITY OF VANE TIP...     35.00  FPS
    VANE STROKE TO HEIGHT RATIO....    0.3250
    VANE STROKE TO THICKNESS RATIO.    1.6000
    MINIMUM VANE THICKNESS.........    0.0400  INCHES
    VANE MATERIAL DENSITY..........    0.0690  LBS/CU IN
    COEF. OF FRICTION VANE TIP.....    0.0500
    WINDAGE COEFFICIENT............    1.5600  HP
    VANE STAGE PRESSURE RISE.......   1200.00  PSI
    ACCELERATION RATIO RADIAL/ROT..      0.60
    PUMP MODEL NUMBER..............SES




OUTPUT DATA
    NUMBER OF VANES................      12.0
    TRACK RING DIAMETER-MAX........    1.6042  INCHES
    VANE STROKE....................    0.0723  INCHES
    VANE THICKNESS-CALC............    0.0451  INCHES
    VANE HEIGHT....................    0.2233  INCHES
    VANE THICKNESS-USED............    0.0451  INCHES
    VANE BEARING LOAD-TIP..........    604.53  PSI
    VANE LENGTH....................    0.6470  INCHES
    VANE LENGTH TO DIAMETER RATIO..      0.44
    VANE ROOT DIAMETER-MAX.........     1.012  INCHES
    TIP DRAG HORSEPOWER LOSS.......    0.1507  HP
    WINDAGE HORSEPOWER LOSS........    0.0762  HP
    TOTAL HORSEPOWER LOSS..........    0.2269  HP
    VANE BENDING STRESS............    7656.7  PSI
    VANE HERTZ STRESS..............  15362.27  PSI
```

Performance

Although the peak overall efficiency of the vane pump is similar to that of a gear pump, the predicted performance characteristics are not similar.

Efficiency

The overall and volumetric efficiency for the steam engine pump is predicted in the following figure. The volumetric efficiency at low speeds is higher than that demonstrated for a gear pump (Figure C-4) which was useful in conserving power for the larger speed range encountered in the steam engine system.

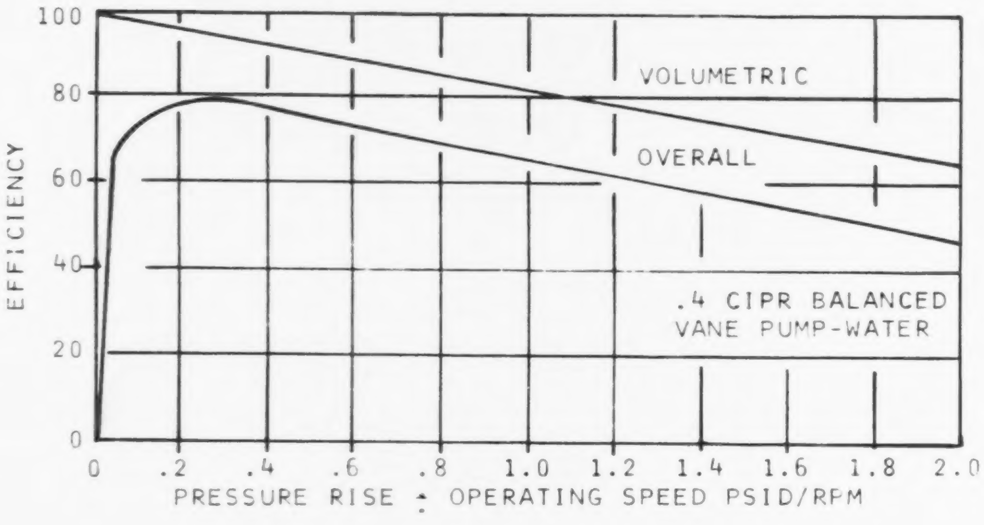


FIGURE D-1

APPENDIX E

INDUCERS

Requirements

The inducer requirements to assure proper charging of the positive displacement pump for each of the Rankine Cycle Systems must be established. Tabulated below is the pump input speed and resulting positive displacement pump tip speed. This parameter defines the pressure rise required from the centrifugal stage which, combined with the flow, establishes the Specific Speed Parameter. Finally, the system inlet pressure above vapor pressure combines with the aforementioned flow and speed conditions to establish the Suction Specific Speed Parameter. These two parameters, Specific Speed and Suction Specific Speed, define the type and degree of complexity of the centrifugal pump design.

System	RPM	Pos Displ Tip Speed fps	Flow Rate gpm	Charging Head ft.	Specific Speed	NPSH ft.	Suction Specific Speed
Steam	4000	35	2.5	38	412	1.188	5,558
Organic Turbine	8000	62.8	28	122	1150	8.317	3,653
Organic Reciprocator	6660	46.5	17	67	1170	4.27	9,244

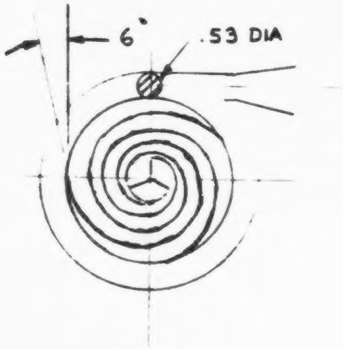
From this analysis, it appears that all cycles have manageable conditions as far as suction conditions are concerned. The NPSH was determined by the physical head in the installation plus the pressure above vapor pressure at the minimum subcooling condenser conditions.

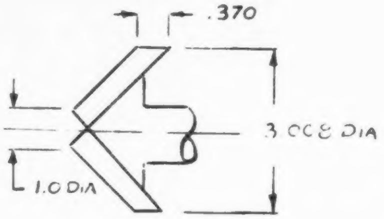
Centrifugal inducers were sized for each of the three applications. As the following descriptions will show, they all are a nominal three inches outside diameter and, therefore, consume the same envelope despite the wide variation in operating requirements.

The Organic Turbine inducer is a three-bladed, unshrouded impeller. It is a specialized mixed flow design which has a cone for the hub and blade tip configurations. The blades, themselves, are perpendicular to the hub surface. The unshrouded design minimizes axial loads due to unequal pressure gradients on the hub or tip shroud. Also, close running labyrinth seals are eliminated.

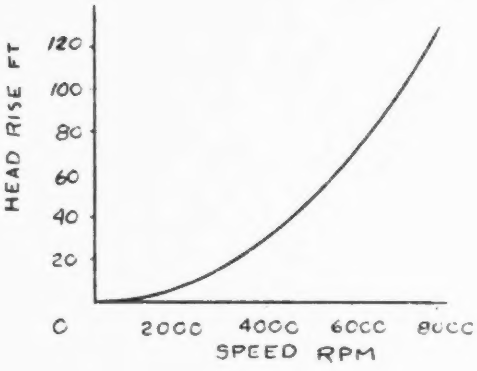
This impeller discharges into a constant area toroidal collector. Such a collector has better efficiency at part flow than a conventional volute or vane type diffuser. It has no circumferential pressure gradient around the impeller and, therefore, no radial loads are imposed on the bearings.

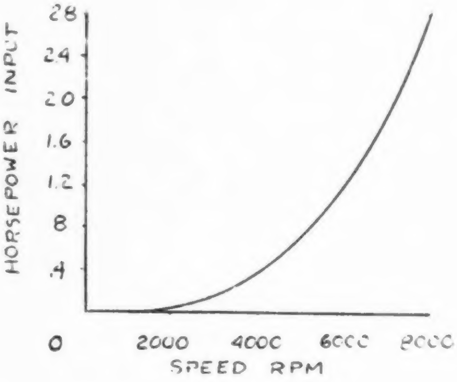
The performance and detail geometric proportions of this centrifugal inducer are described in the following set of figures.





ORGANIC TURBINE SYSTEM FEED PUMP INDUCER

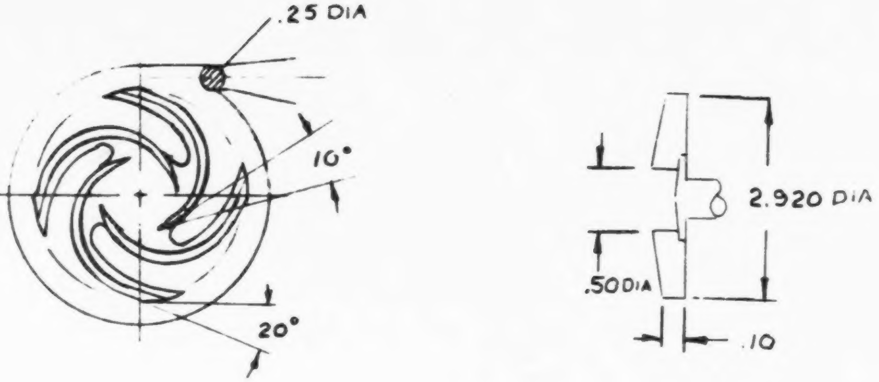




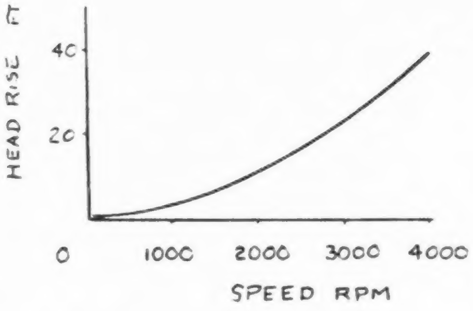
This centrifugal inducer would also satisfy the requirements of the organic reciprocator system with a little margin. Actually, the outside diameter could be reduced to 2.7 inches and still adequately charge the positive displacement section of the organic reciprocator boiler feed pump.

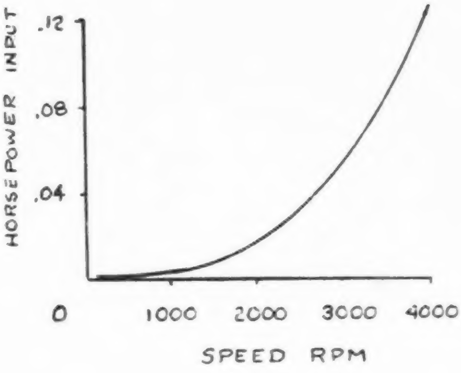
This application also has the possibility of being satisfied by the same centrifugal inducer sized for the steam system. This inducer consists of a four-bladed, unshrouded radial impeller which discharges into a constant area toroidal collector.

A radial impeller was selected because it offers the best overall efficiency in this specific speed range while still satisfying the suction requirements. The reasoning for the other geometrical considerations is the same as presented for the Organic Turbine System. The performance and detail geometric proportions of this inducer are described in the following figures.



STEAM SYSTEM FEED PUMP INDUCER

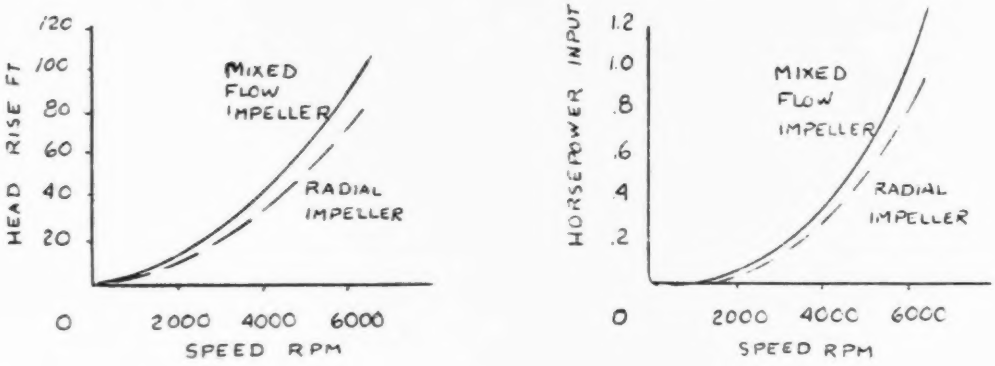




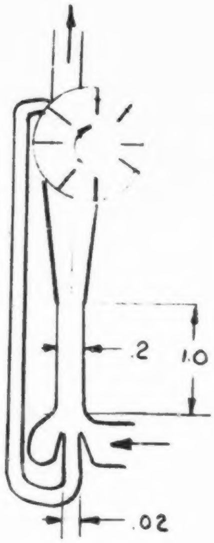
The performance of either of these inducers, when applied to the Organic Reciprocator System, is shown in the following set of figures.

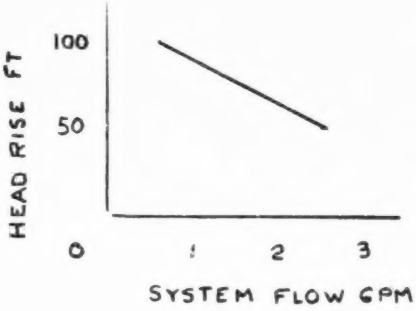
It should be noted that the reported test results, presented in another section of this report, agree with this performance and the table on Page E-1.

ORGANIC RECIPROCATOR SYSTEM FEED PUMP INDUCER



Still another alternative is available for the steam system and that is to employ a jet inducer. This inducer would use bypass flow from the positive displacement pump to induce flow at its own inlet by a venturi effect. Shown below is the schematic of this system, together with the sizing of the inducer and its predicted performance.





This presents obvious potential improvements such as using a pintle nozzle in the jet inducer to fulfill the bypass function for the positive displacement pump.

APPENDIX F

VARIABLE DISPLACEMENT VANE PUMP

Pump Size

The analyses presented in Appendices C and D establish that .9 cubic inches per revolution is the desirable displacement for the organic turbine cycle pump. Chandler Evans has a pump of this capacity, the performance of which is recorded below.

Pump Performance

The following curves show the performance of the MFP-330 variable displacement vane pump in the full stroke position (Figure F-1) as modified by partial stroke operation (Figure F-2). The dotted lines show performance improvements to be attained with low density and hydrodynamic vanes.

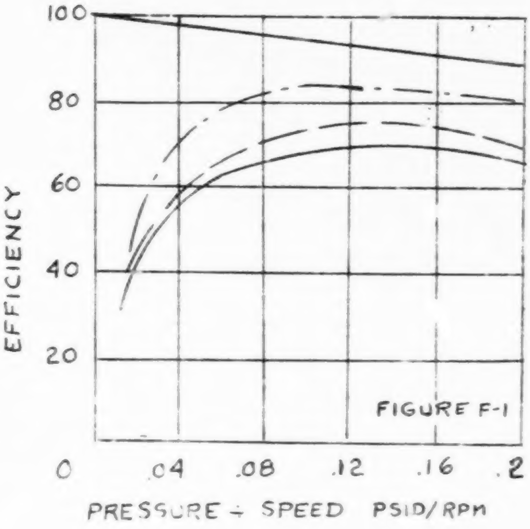


FIGURE F-1

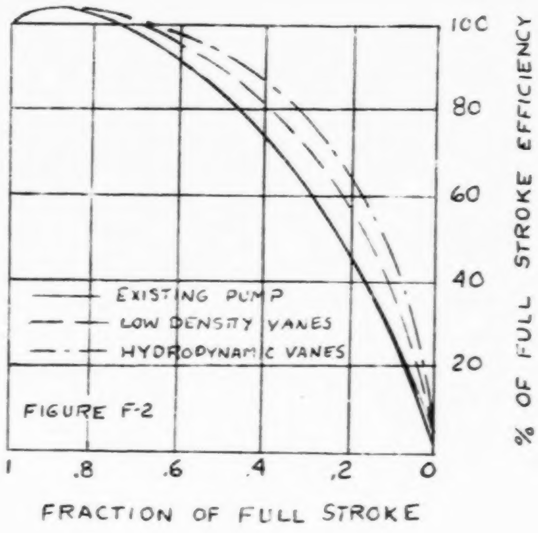


FIGURE F-2

System Performance

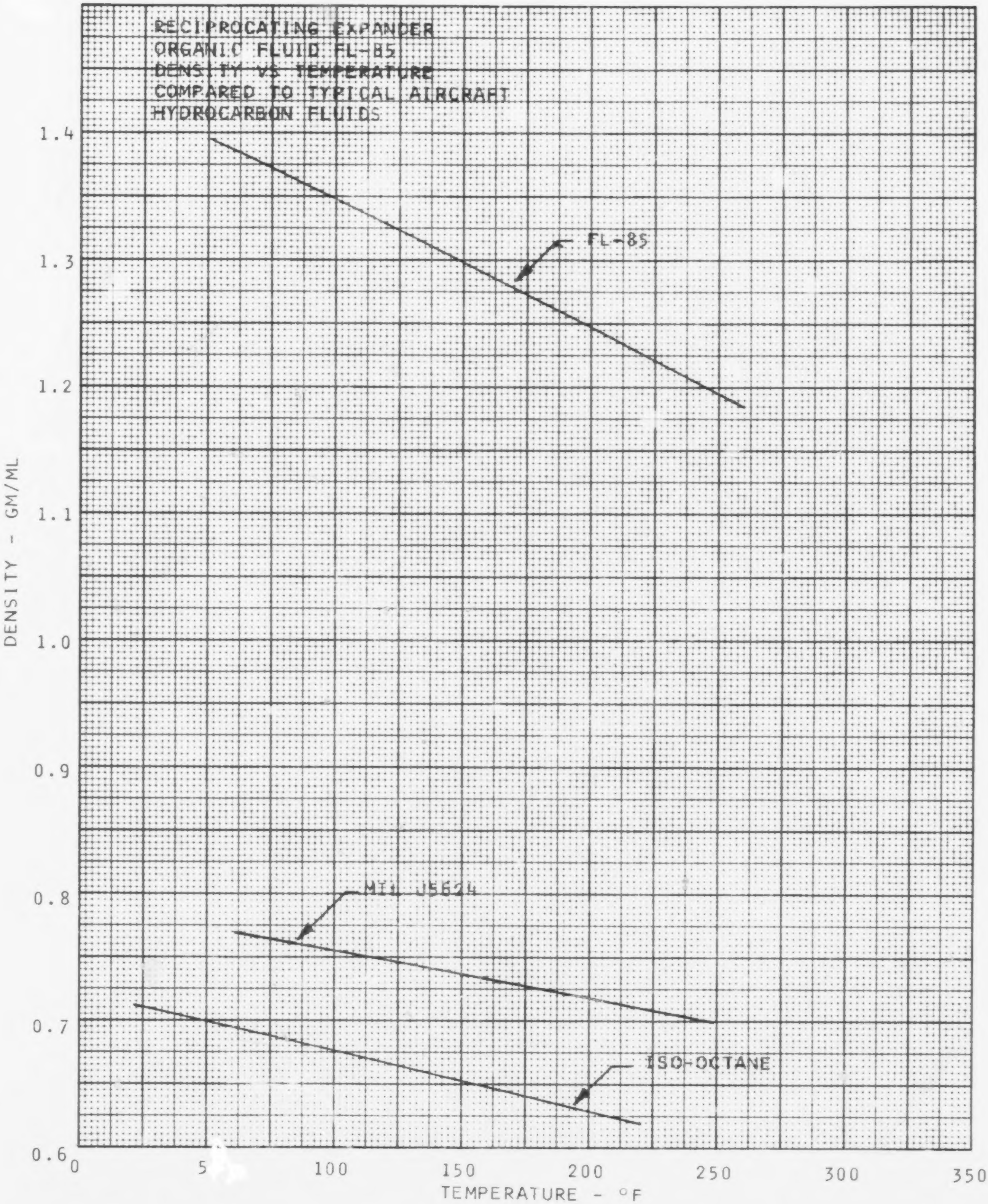
The following table presents the predicted variable displacement pump system performance for the organic turbine cycle engine.

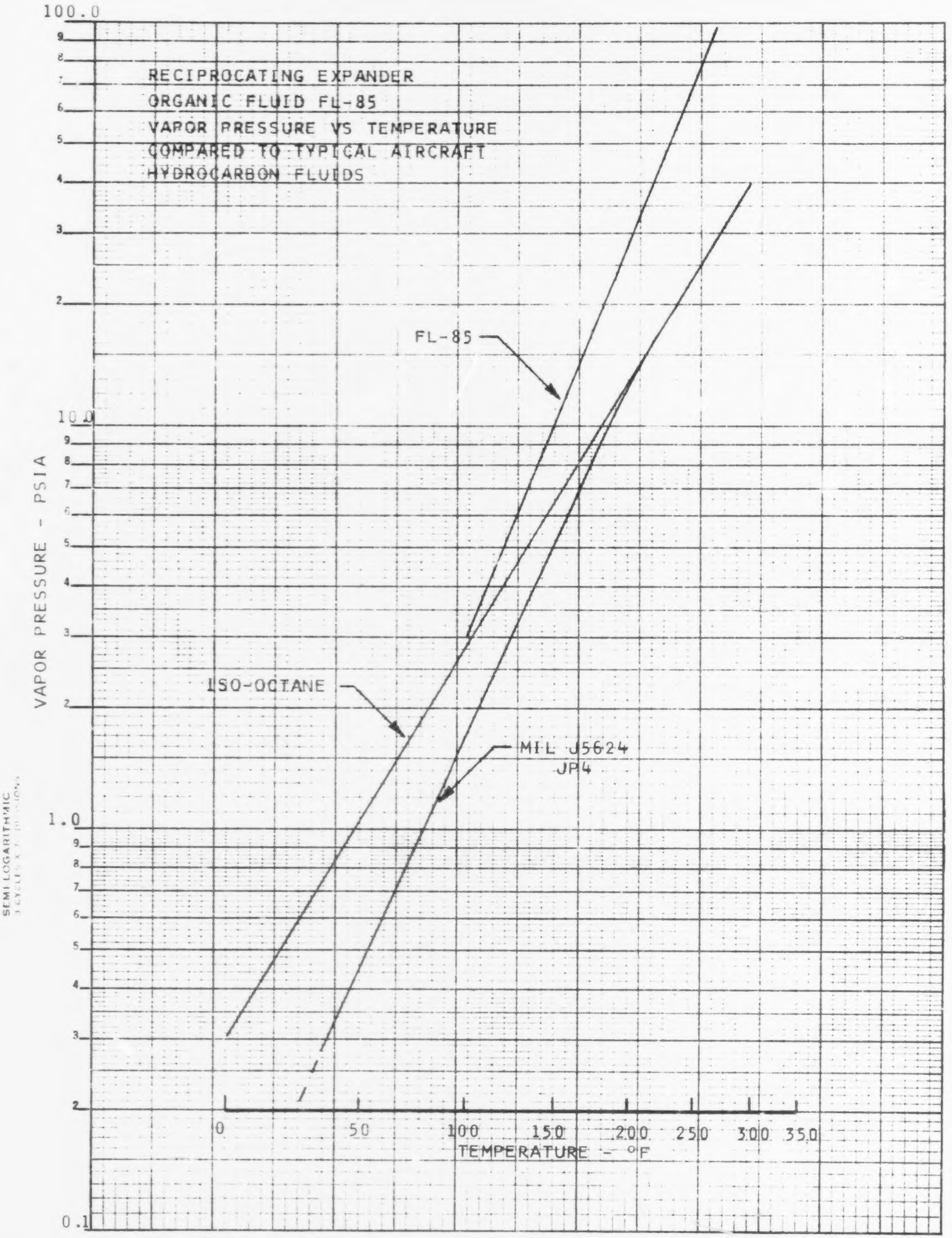
18 gpm Idle Speed Pump

Operating Point	Pump Speed RPM	Pressure Rise PSI	System Flow GPM	Maximum Pump Flow GPM	Existing Pump System Efficiency	Low Density Vanes	Hydro-dynamic Vanes
1	5256	1050	1.8	18.4	22	25	36
2	5256	1050	20	18.4	67	74	86
3	7500	1050	28	26.3	71	77	87
4	9756	1050	28	34.3	66	71	83
5	9756	1050	2	34.3	19	19	25
1'	5256	600	1.8	19.5	20	33	38
2'	5256	600	20	19.5	68	74	86
3'	7500	600	28	28	65	70	82
4'	9756	600	28	36.2	60	65	78
5'	9756	600	2	34.3	18	19	23

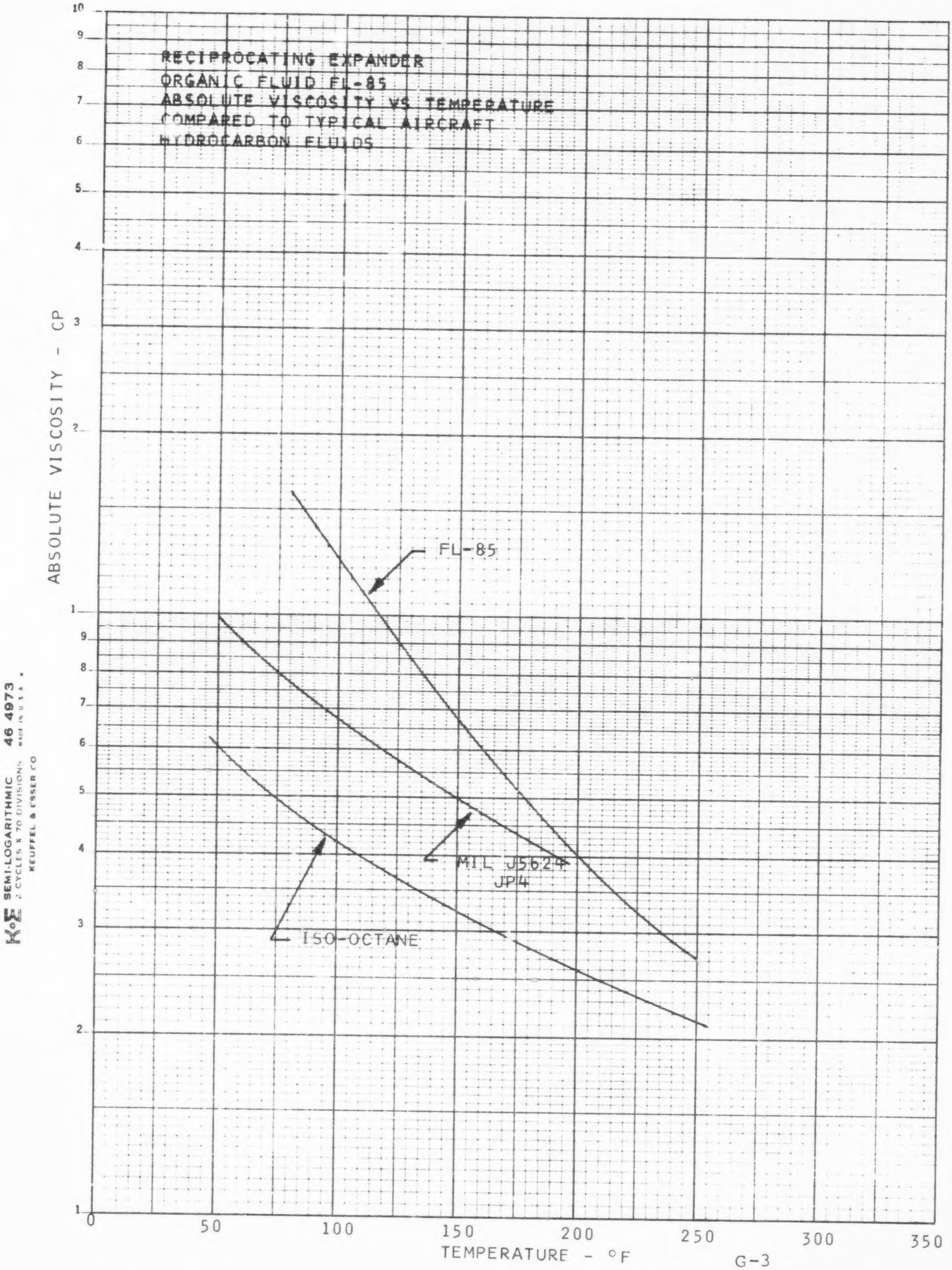
APPENDIX G

Cycle Fluid Properties and Comparisons

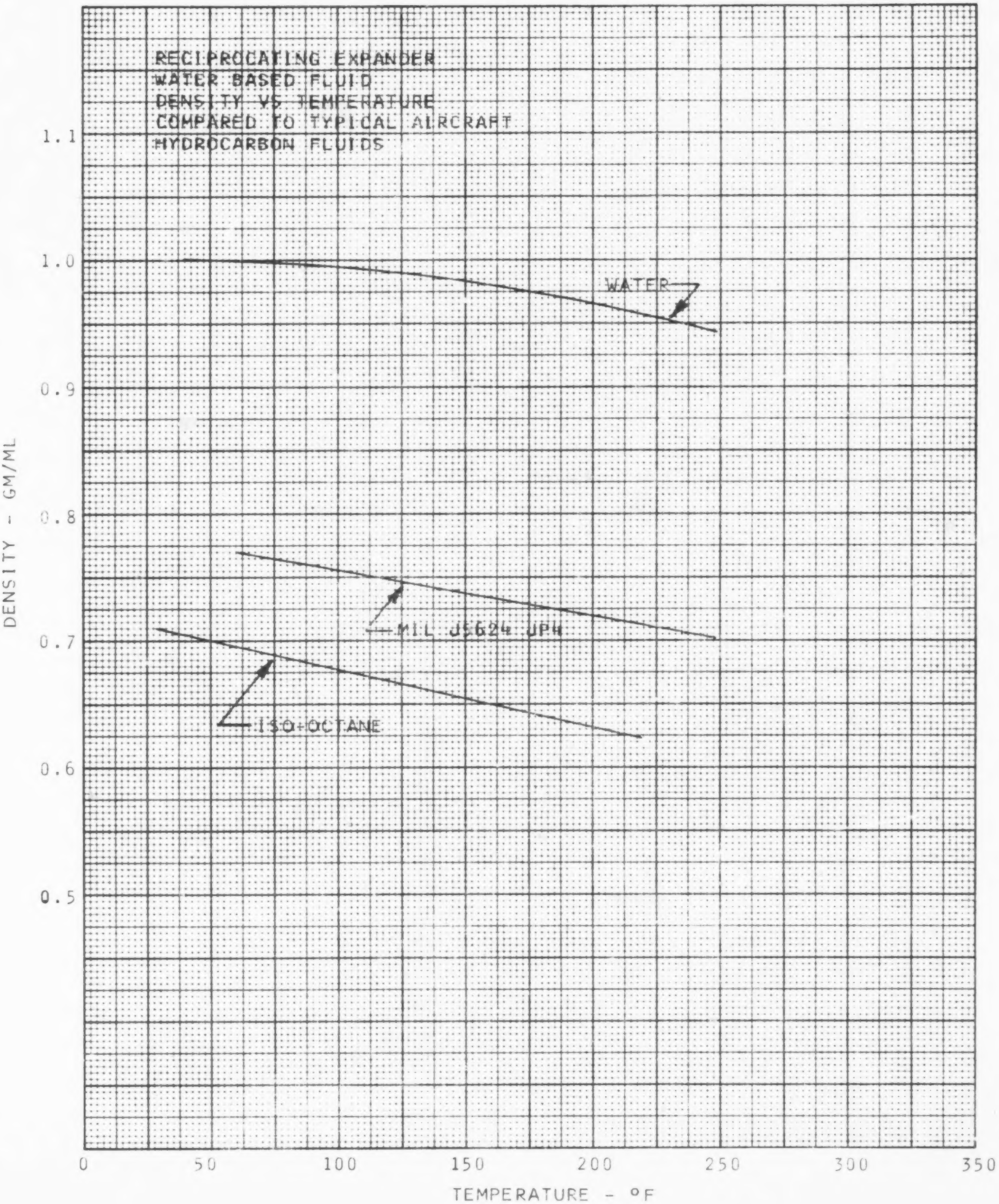

RECIPROCATING EXPANDER
ORGANIC FLUID FL-85
DENSITY VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS
FL-85
MIL J5624
ISO-OCTANE
DENSITY - GM/ML
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0
50
100
150
200
250
300
350
TEMPERATURE - °F

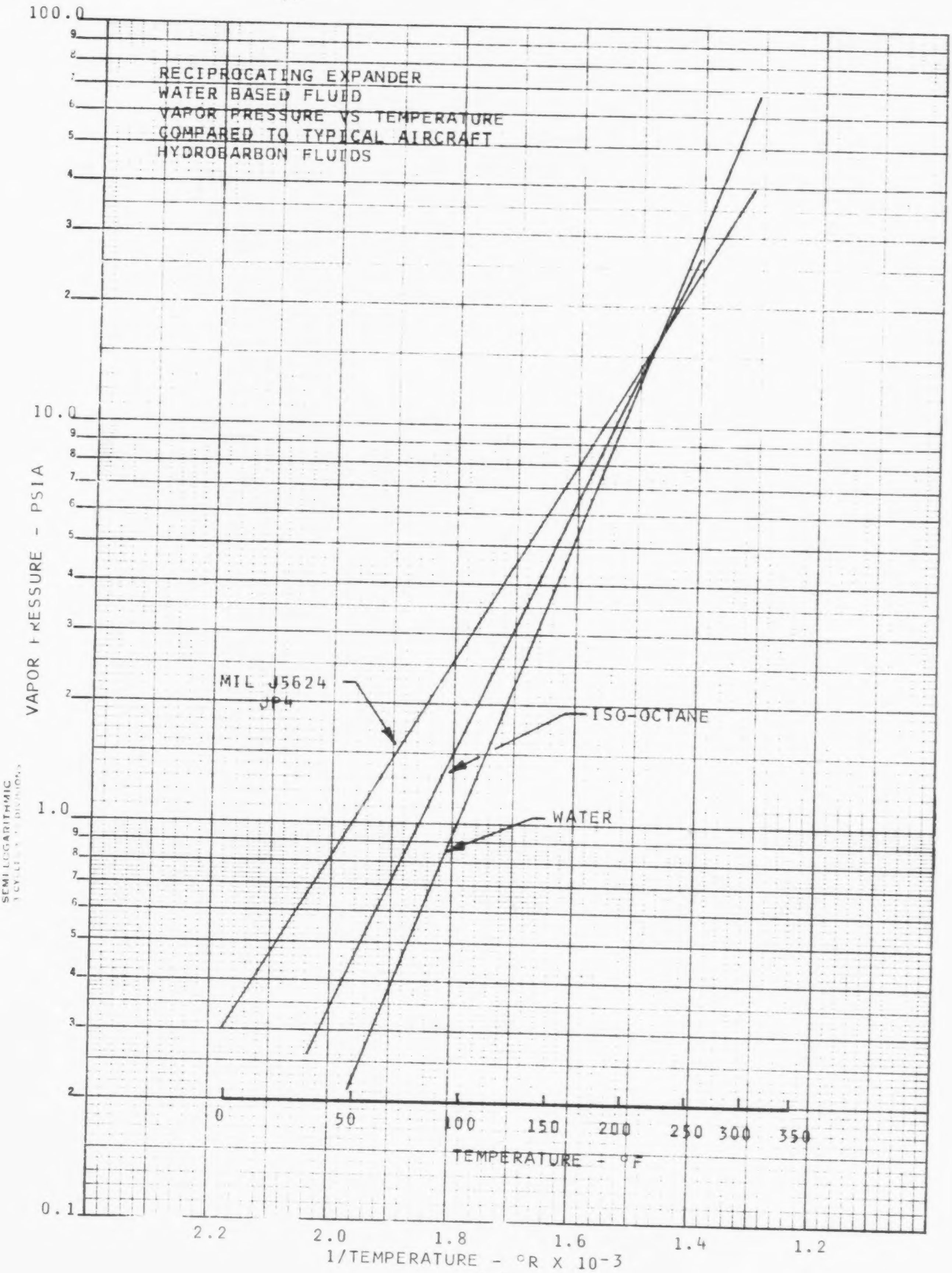


RECIPROCATING EXPANDER
ORGANIC FLUID FL-85
VAPOR PRESSURE VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS

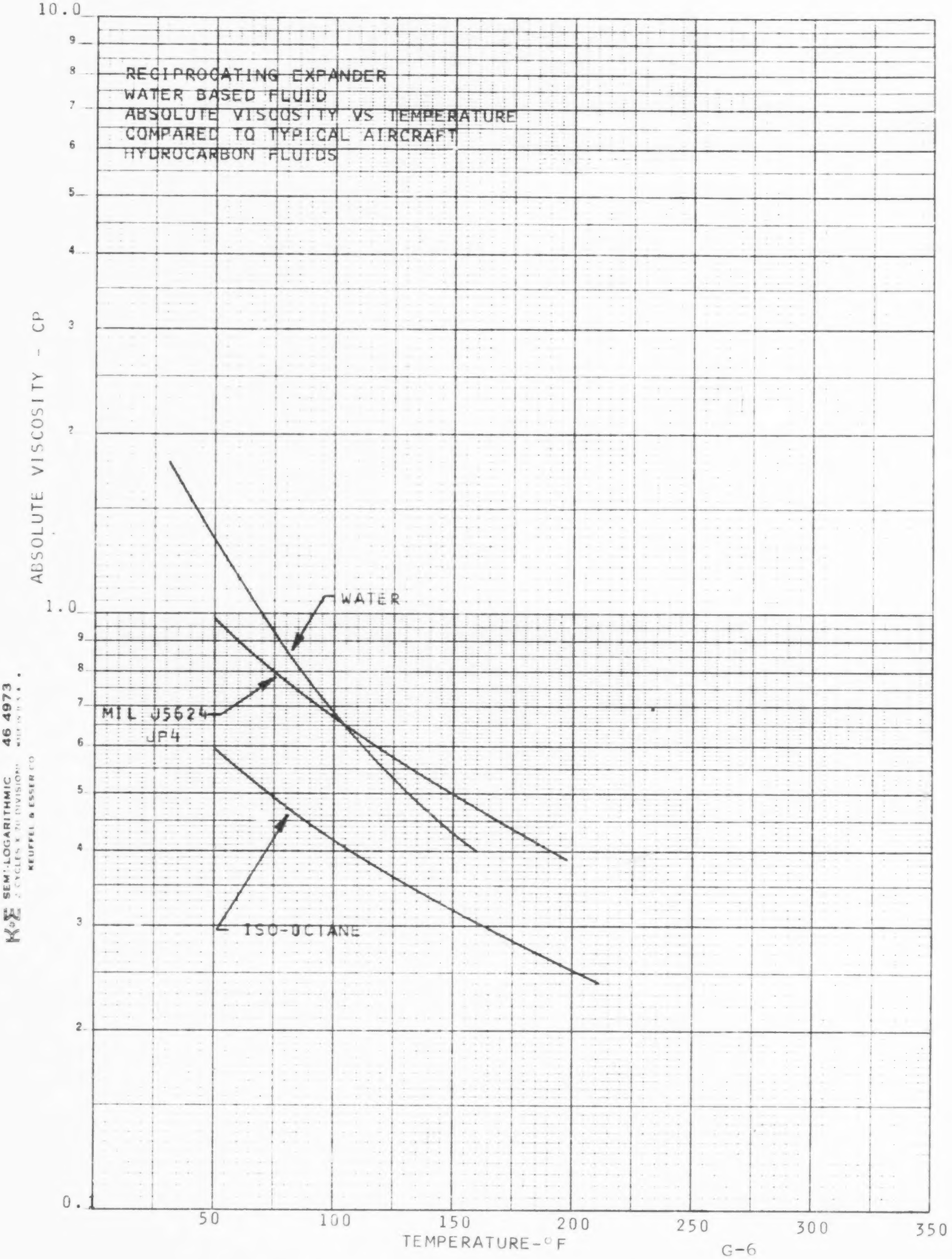


RECIPROCATING EXPANDER
ORGANIC FLUID FL-85
ABSOLUTE VISCOSITY VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS

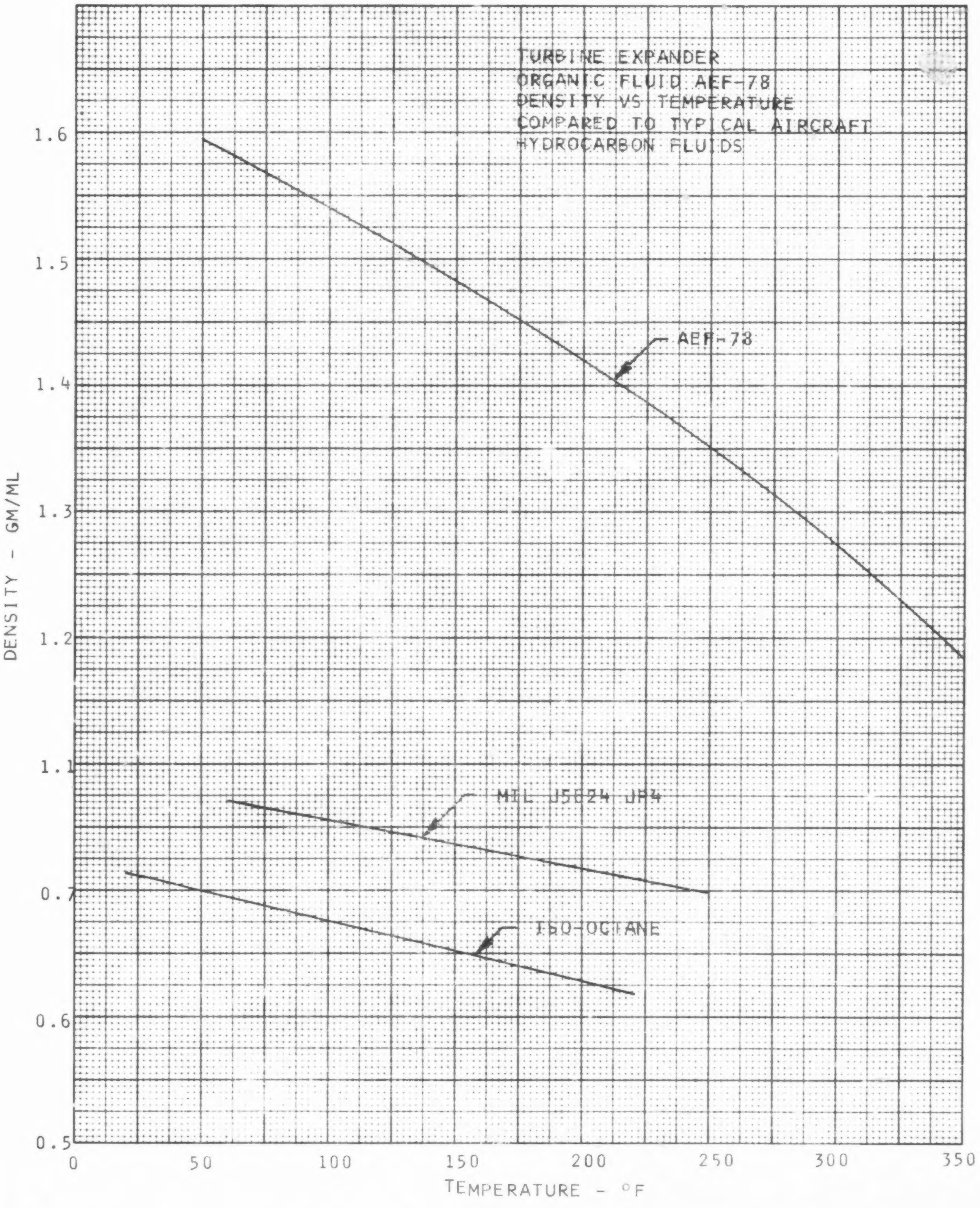

RECIPROCATING EXPANDER
WATER BASED FLUID
DENSITY VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS
WATER
MIL J5624 JP4
ISO-OCTANE
DENSITY - GM/ML
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0
50
100
150
200
250
300
350
TEMPERATURE - °F

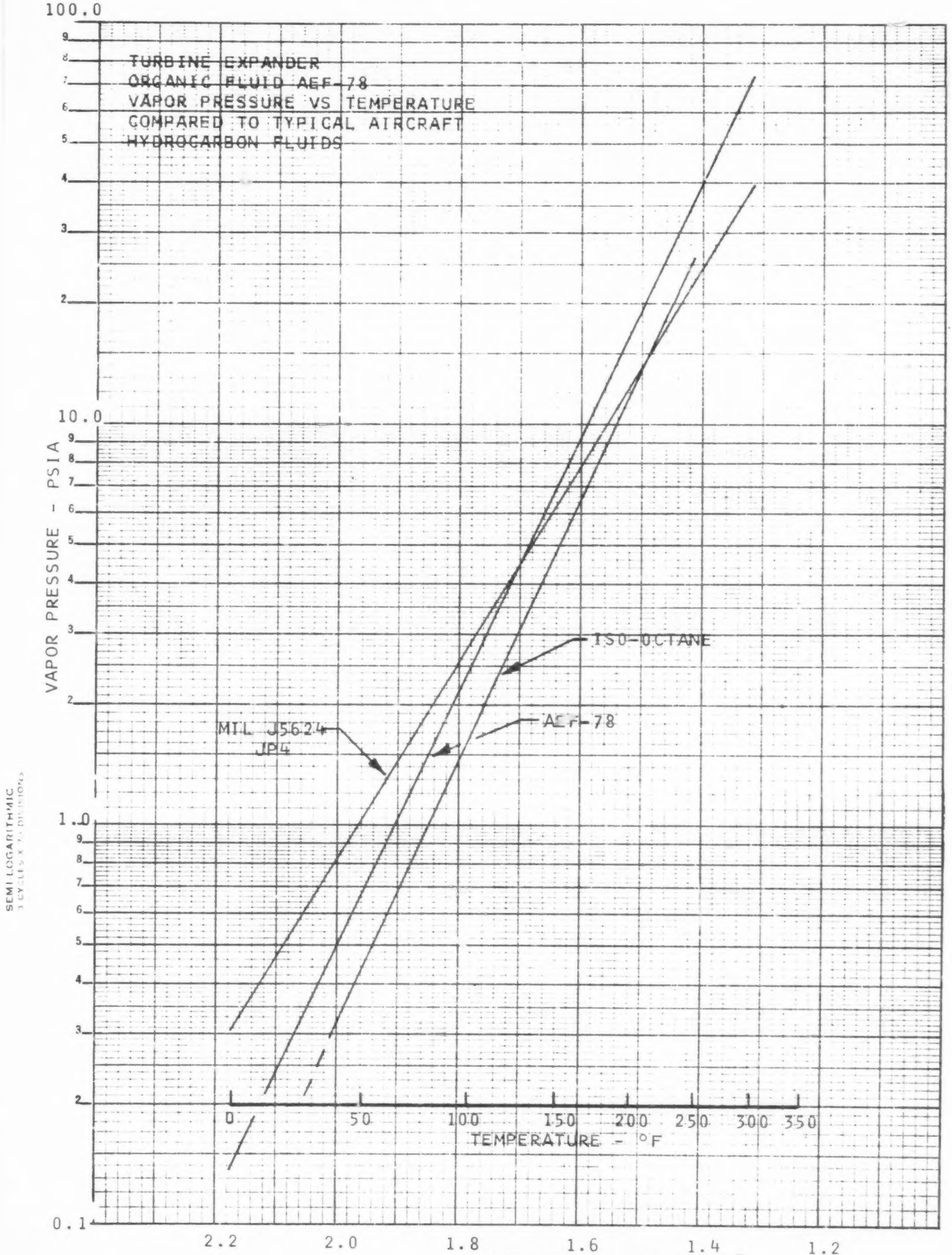


RECIPROCATING EXPANDER
WATER BASED FLUID
VAPOR PRESSURE VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS

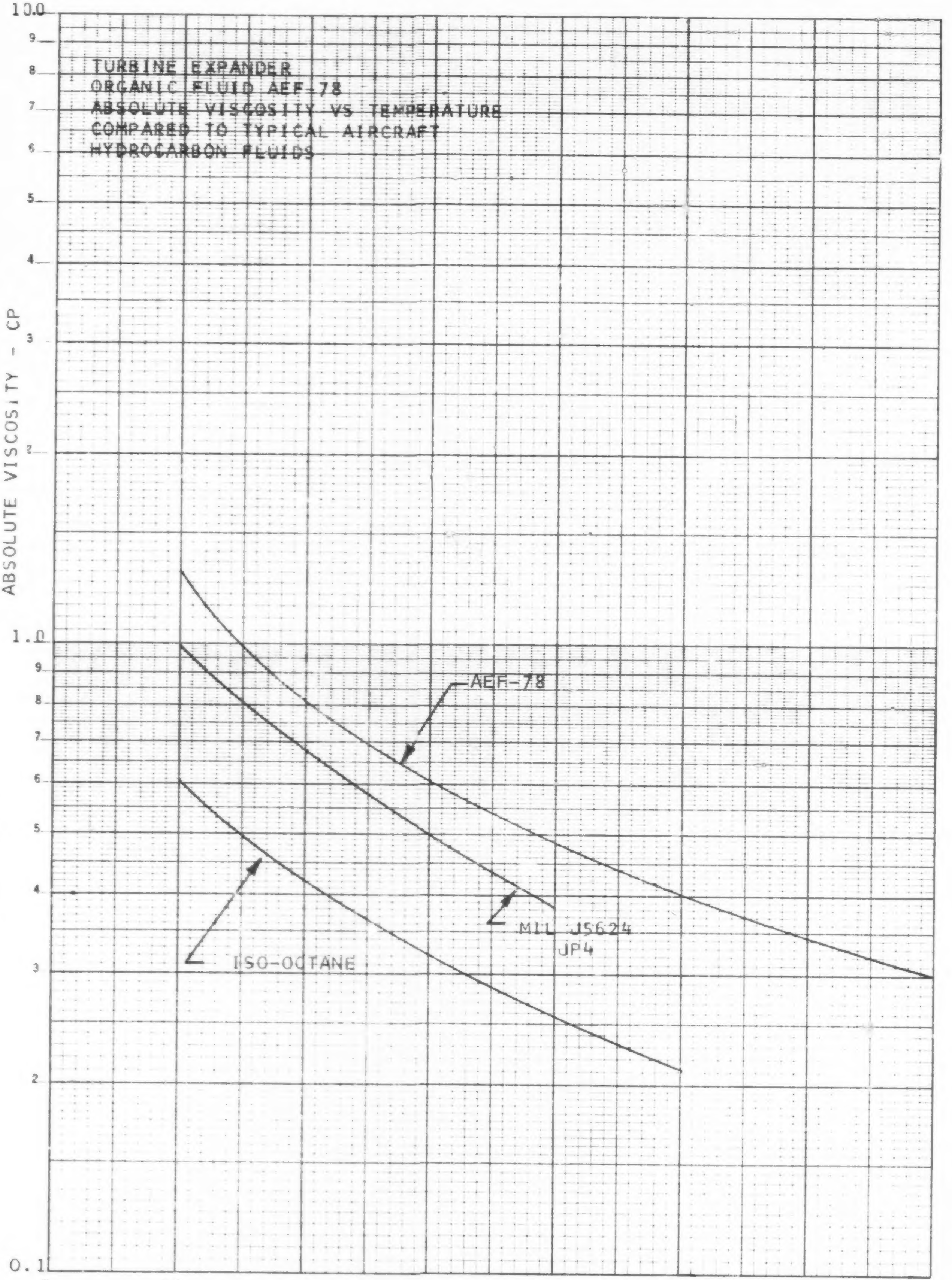


RECIPROCATING EXPANDER
WATER BASED FLUID
ABSOLUTE VISCOSITY VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS

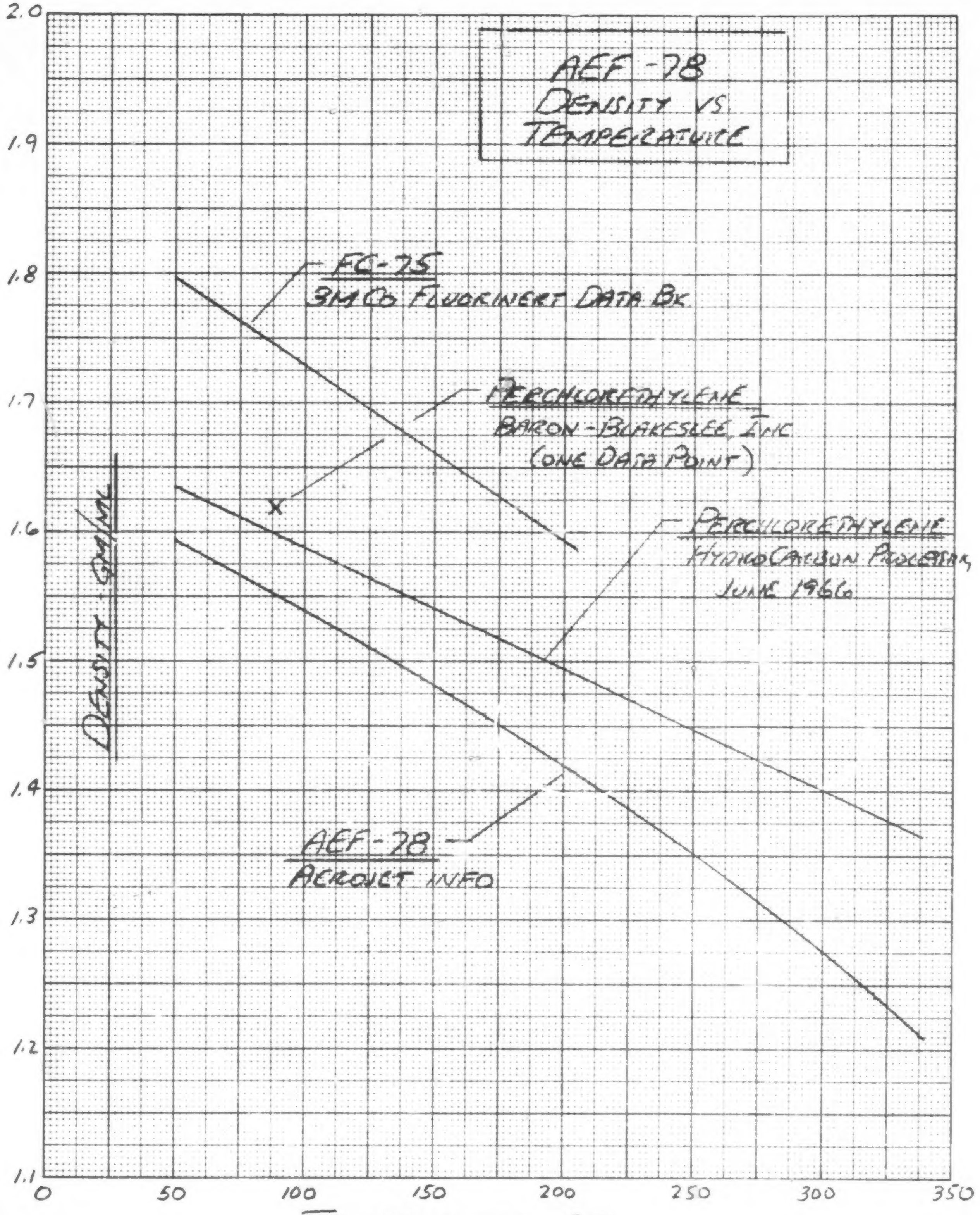

TURBINE EXPANDER
ORGANIC FLUID AEF-78
DENSITY VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS
AEF-78
MIL J5624 JP4
ISO-OCTANE
DENSITY - GM/ML
1.6
1.5
1.4
1.3
1.2
1.1
0.7
0.6
0.5
0
50
100
150
200
250
300
350
TEMPERATURE - °F

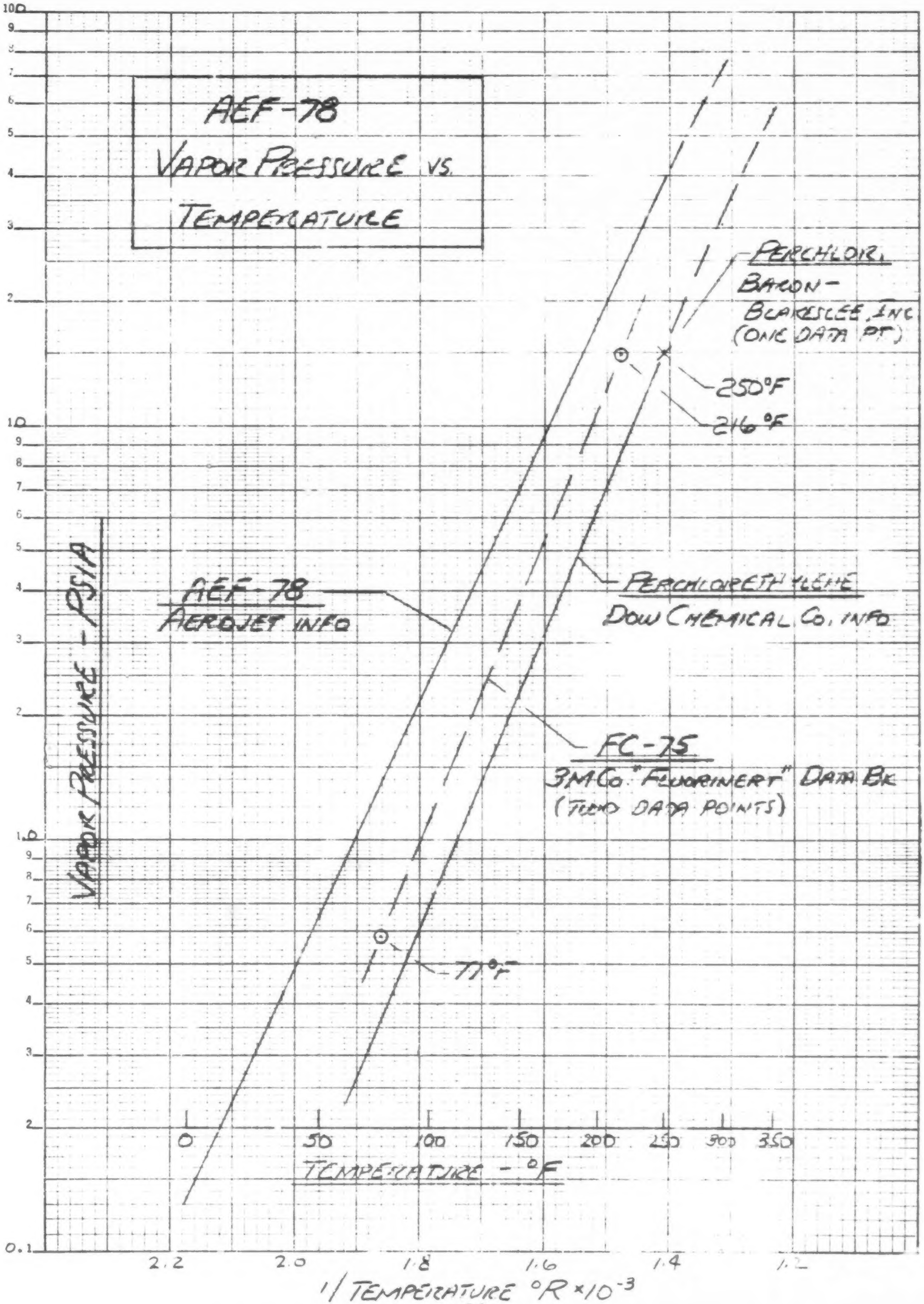


TURBINE EXPANDER
ORGANIC FLUID AEF-78
VAPOR PRESSURE VS TEMPERATURE
COMPARED TO TYPICAL AIRCRAFT
HYDROCARBON FLUIDS

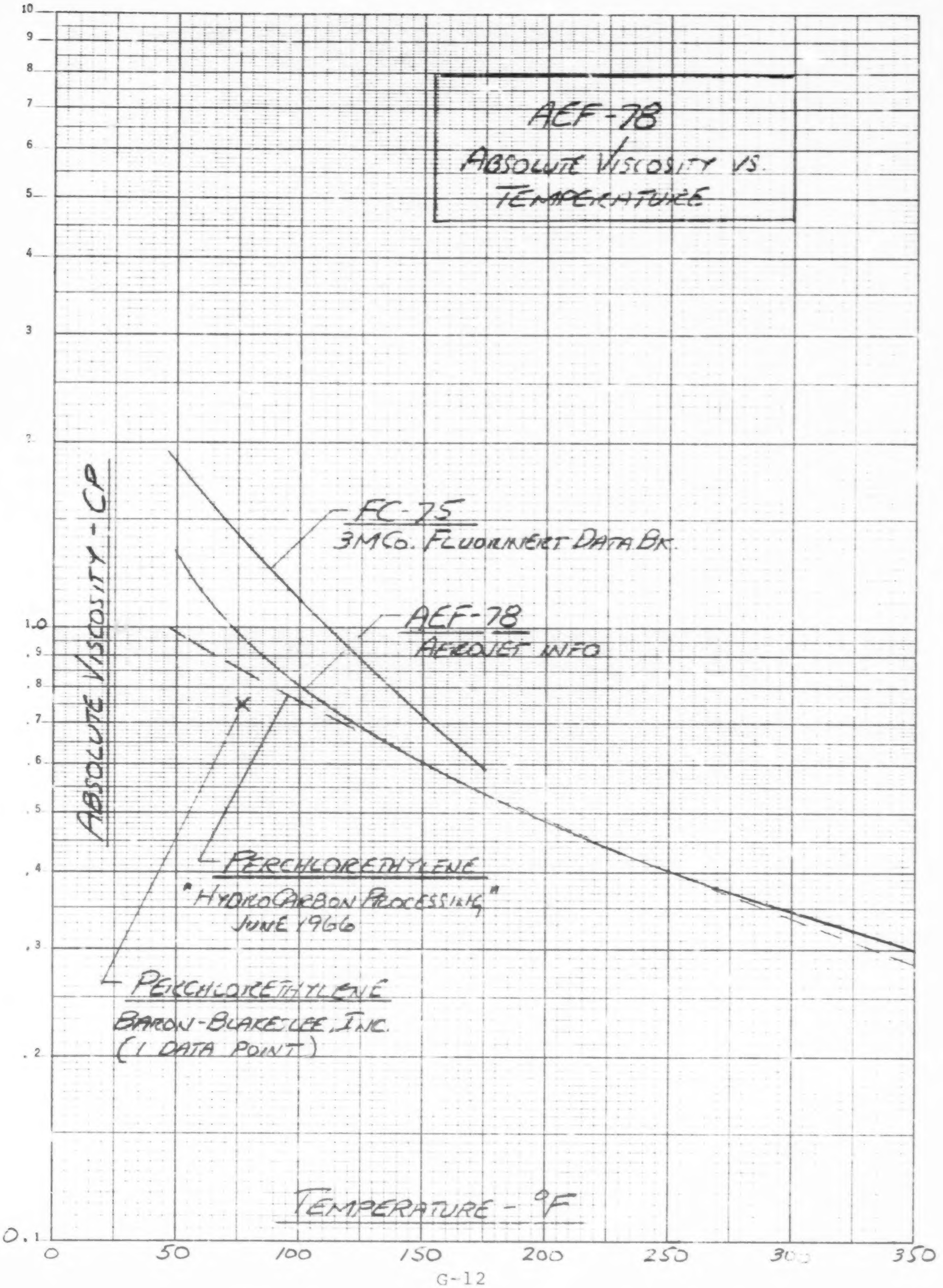


K+E SEMI-LOGARITHMIC 46 4973
2 CYCLES X 70 DIVISIONS MADE IN U.S.A.
KEUFFEL & ESSER CO.


AEF-78
DENSITY VS.
TEMPERATURE
FC-75
3M Co FLUORINERT DATA BK
PERCHLORETHYLENE
BARON-BLAKESLEE INC
(ONE DATA POINT)
PERCHLORETHYLENE
HYDROCARBON PROCESSING,
JUNE 1966
AEF-78
AEROJET INFO
DENSITY GM/ML
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
0
50
100
150
200
250
300
350
TEMPERATURE - °F



SEMI LOGARITHMIC



K&E SEMI-LOGARITHMIC 46 4973
2 CYCLES X 70 DIVISIONS MADE IN U.S.A.
KEUFFEL & ESSER CO.

APPENDIX H

Piston Pumps

The parametric study of piston pumps operating on low viscosity fluids, from which the curves in this section were derived, proceeded from a review of actual performance data acquired at Chandler Evans and its English licensee, Dowty Ltd..

The development by Thermo Electron of a variable displacement piston pump for the Rankine cycle automotive application was also included as input to this investigation. The TECO data was obtained from their published reports.

Most of the pumps investigated were of radial piston configuration operating on aircraft fuels having viscosities around 1-2 centistokes.

Efficiency and net positive suction head as a function of pump displacement were the parameters of prime concern in this study. The following conclusions were drawn:

1. Efficiency

Manufacturing tolerances and operating clearances, which contribute to leakage losses, vary little with pump size. For this reason, the peak overall efficiency of a piston pump increases with displacement.

Windage and porting losses have a more direct relationship with pump displacement.

The net effect of all of these losses tends to cause the peak overall efficiency to become asymptotic to the 90 percent line as size increases. This is shown in Figure H-1.

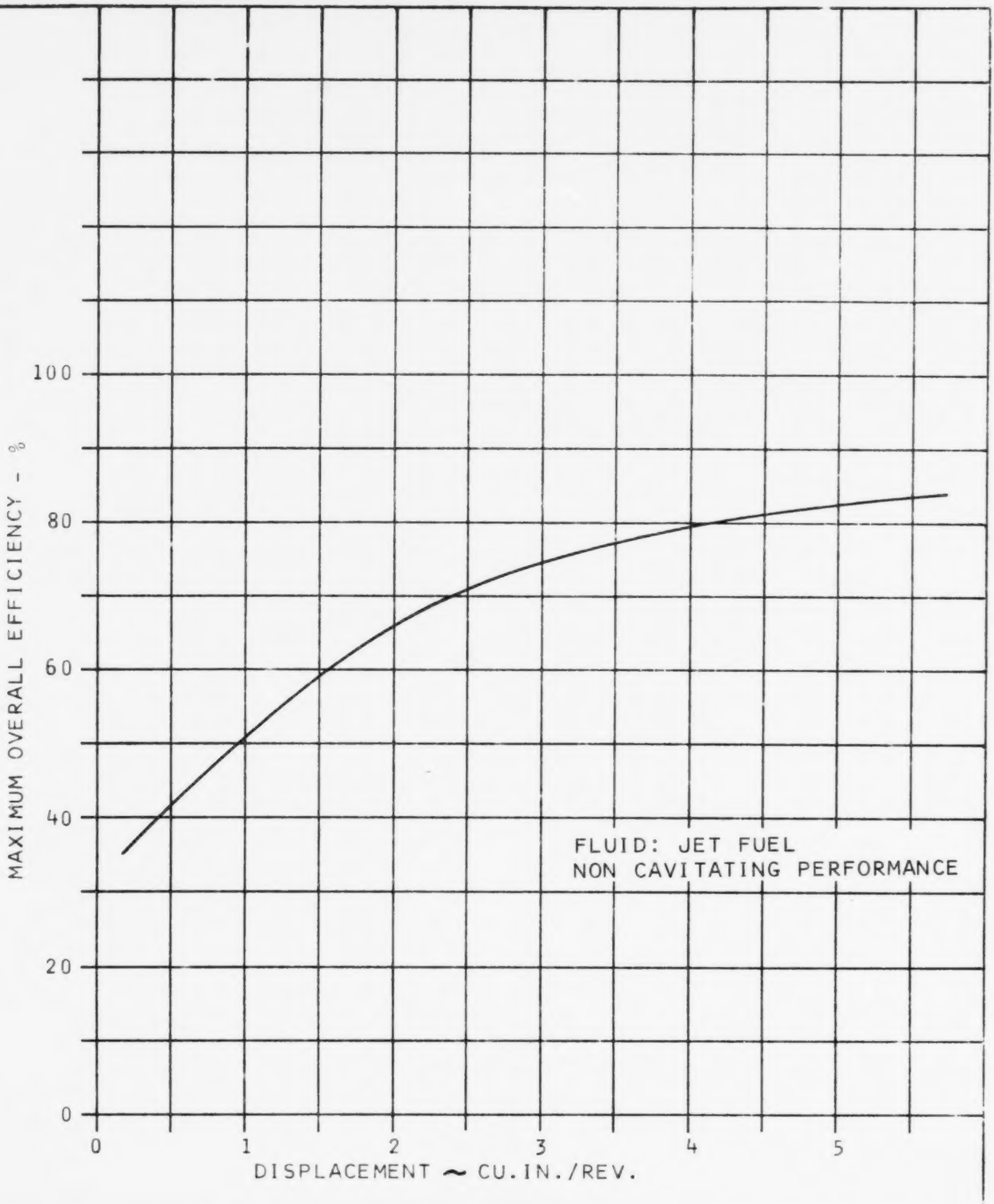
Figure H-2 depicts both volumetric and overall efficiencies as functions of a normalized pressure parameter for a piston pump of 2.3 cubic inches per revolution (CIPR) displacement.

Figure H-3 indicates the value of this normalized pressure parameter at which peak overall efficiency occurs, with respect to displacement.

All of this points to a large, slow piston pump as being more efficient.

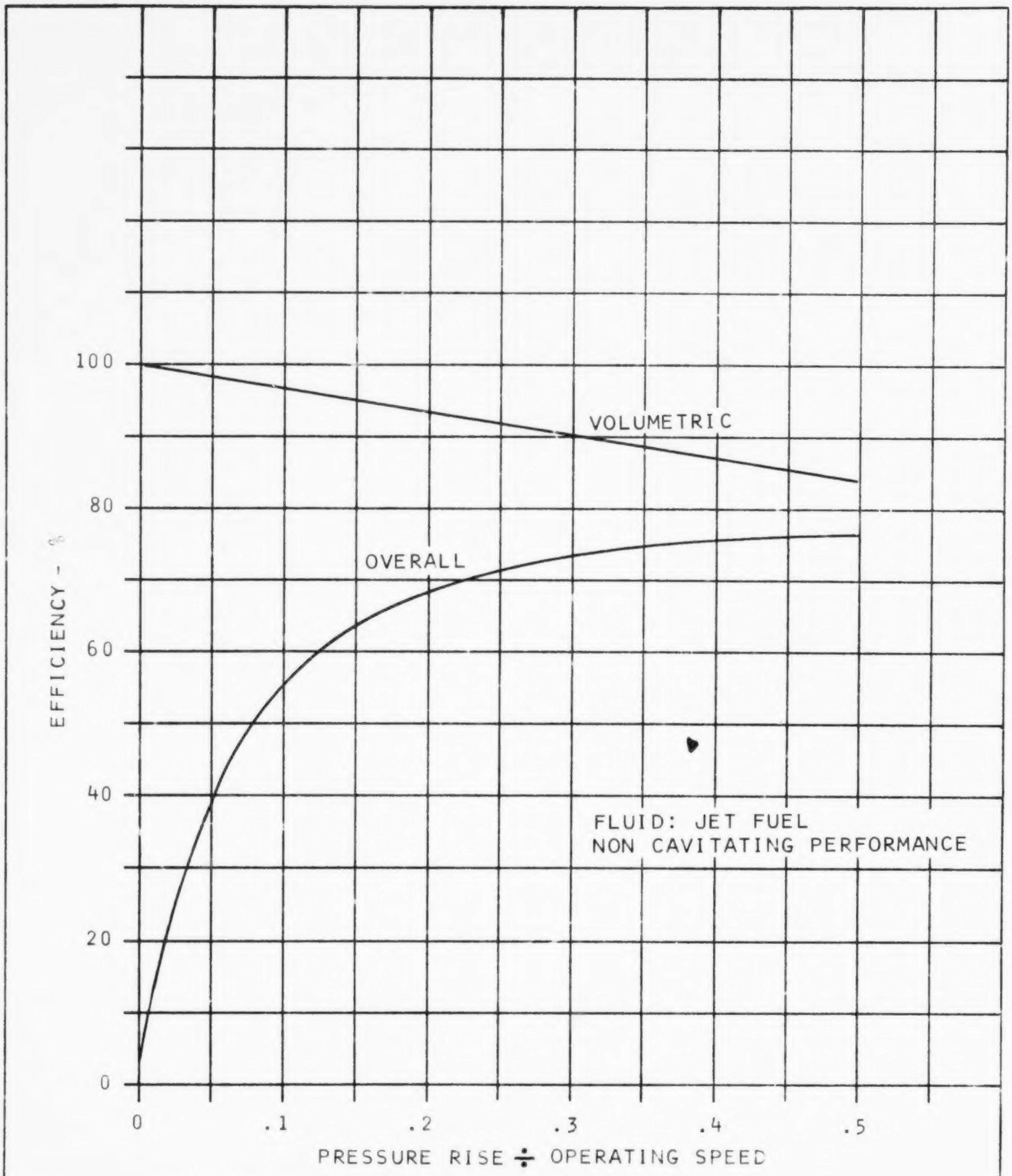
2. Net Positive Suction Head

Inlet pressure requirements for piston pumps can be determined from Figure H-4. If spring loaded valves are used, the pressure parameter will follow a steeper curve than shown. It can be seen that a small, slow pump needs less inlet head for satisfactory performance.



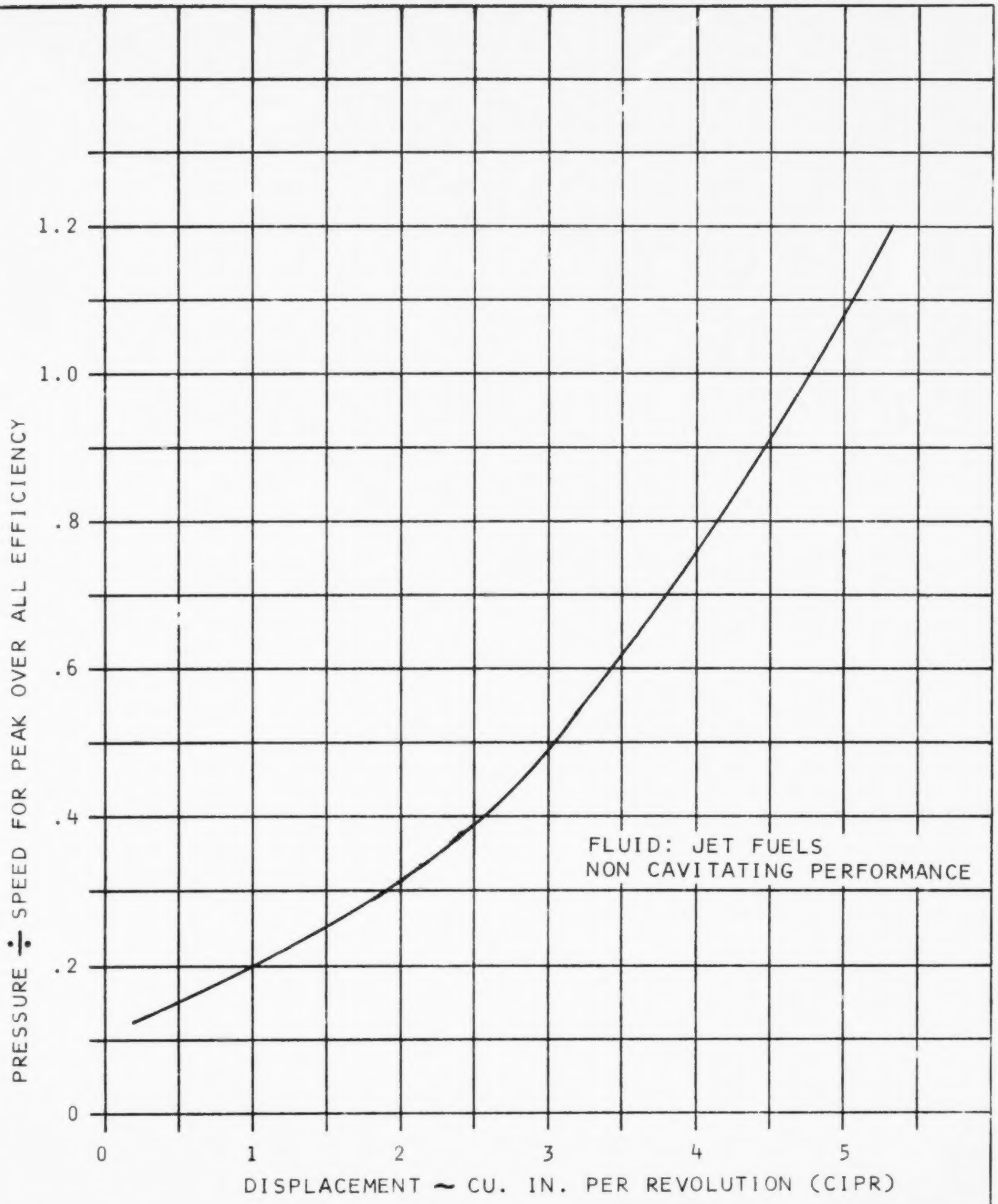
ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** PISTON PUMP EFFICIENCY VS DISPLACEMENT
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** H-1

Colt Industries Inc Chandler Evans Inc Control Systems Division

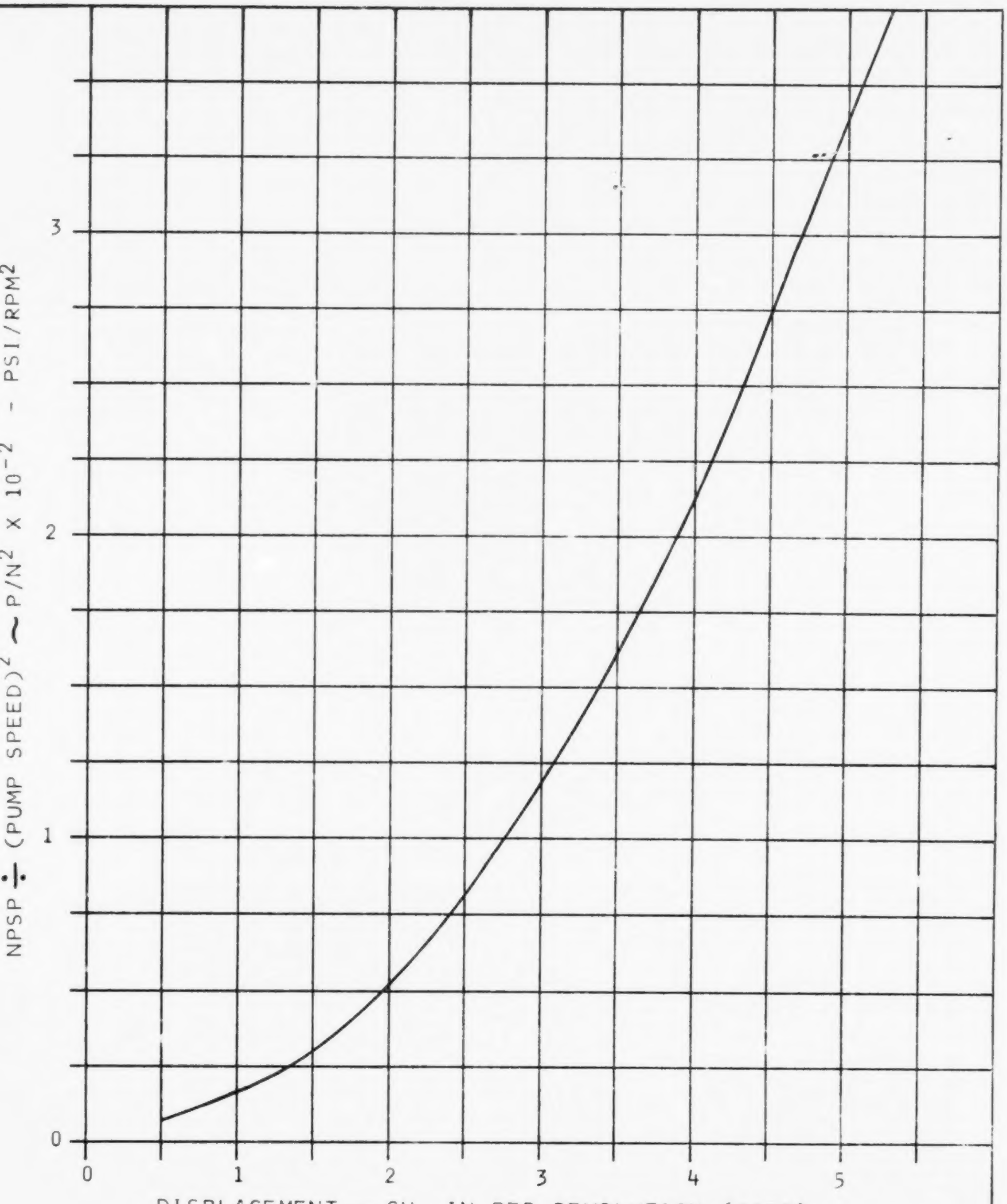


ENG. REPORT NO. APTD-1357	**REPORT TITLE**	**CURVE TITLE** EFFICIENCY VS ΔP/N FOR PISTON PUMP 2.3 CIPR.
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** H-2

Colt Industries Inc **Chandler Evans Inc Control Systems Division**



ENG. REPORT NO.	REPORT TITLE	CURVE TITLE
APTD-1357		ΔP/N VS CIPR FOR PISTON PUMPS AT PEAK OVERALL EFFICIENCY
DATE 12-5-72		
PREPARED BY		FIGURE NO. H-3



ENG. REPORT NO.	REPORT TITLE	CURVE TITLE
APTD-1357		NPSP/N^2 VS CIPR FOR PISTON PUMPS
DATE 12-5-72		
PREPARED BY		**FIGURE NO.** H-4

APPENDIX I

Two-Stage Pumping

The high overall pump efficiencies specified by Contract No. 68-01-0430 tax the performance capability of the ordinary positive displacement pump, especially at low speeds. Adding to the problem are the inefficiencies accruing to a variable delivery pump, either running at part displacement or operating on a variable speed drive.

Because of engine demands, the work performed by the feed pump is necessarily high, especially in the ALRC system. Efficient pump operation contributes substantially to economic engine performance.

The variable delivery mode of operation in some of the pumping systems analyzed in this project is accomplished by a fixed displacement pump on a variable speed drive. Toward optimizing the efficiencies of this combination, consider the following:

1. Limit the turndown range of the variable speed drive to stay at or near maximum drive efficiency.

2. Limit the turndown range of the pump to stay at or near maximum pump efficiency. Accumulators may be used to handle short-term transients and bypass valves to handle situations below steady-stage running conditions.

3. Since in normal vehicle operation, the feed pump output is in the low flow fange, devise a pumping system which calls in full capacity only at high demand. The ALRC system requirements displayed in Figure I-1 show the logic of this approach.

Item 3, above, can be achieved by a variable displacement pump, however, especially in the case of a vane pump, the efficiency penalty from operating at part displacement is too severe. Accomplishing this with fixed displacement pumps produces, in essence, a two-displacement pump. This can be done with any type of pump.

A review of Appendices C and D suggests a method of "staging" pumping elements to optimize efficiency.

If the discharge of a pumping element is directed to the inlet of another equal or slightly smaller element, the output is that of a single element and each supports only half of the total pressure rise. Figure I-2 depicts a schematic of such an arrangement.

When the demand exceeds some predetermined level, the flow paths are switched so that each element is pumping independently of the other. In this condition, the full pressure rise is borne by each element.

A two-element pump can thus be operated either in series or in parallel.

The advantage of this technique can be seen by referring to the efficiency versus pressure ÷ speed curve on Page D-5 of Appendix D. Within a given range of displacements, the peak efficiency occurs at a distinct value of this ratio. The peak efficiency can, therefore, be obtained at several points within an operating envelope by varying both speed and pressure rise. This basic information was used in plotting the curves in Figure I-3 for specific speeds and pressures.

It can be seen that by varying the pressure rise across each element, the best efficiency characteristic of the entire pump can be shifted with speed.

As an additional advantage to this method, the speed ratio range of the variable speed drive can be narrowed, which tends to optimize drive efficiency as well.

The efficiency curves for two-stage operation presented in the main body of this report were plotted by following this line of analysis. Installation studies were made, some of which are illustrated in Figures 45, 47 and 49 in the main body of the report.

It thus remains to select the parameters that will control the switching point and to provide a physical means to effect smooth transitions between series and parallel operation.

These problems will be resolved in cooperation with the system contractor while the pump is being integrated into the complete system. However, conjecturing a bit, the following control modes are suggested:

1. Organic Fluid Reciprocator - TECO

   1.1 It was noted in the TECO system that the intersection of the efficiency curves for series and parallel operation varied with expander speed. Since discharge pressure is constant, engine flow is ultimately established by throttle position. The throttle mechanism can also load a spring against which a flyweight actuator may operate. Thus, for each flow rate, switching will occur at a unique and different engine speed.

1.2 A simpler method would establish the switching point strictly by throttle position. There would be some sacrifice of efficiency at certain conditions, but this may be outweighed by the simplicity of the control mechanism.

2. Water Reciprocator - SES

The two-stage pump presented for the water reciprocator was intended to reduce the lost work in a bypassing system. In this application, a venturi flow sensor in the bypass line may be balanced against one in the boiler delivery line so that when bypass flow exceeds engine flow, the pump would switch from parallel to series operation.

3. Organic Fluid Turbine - ALRC

3.1 Since the ideal switching points for the organic fluid turbine are also a function of expander speed, a speed actuator similar to that described for the organic fluid reciprocator may be considered.

3.2 Throttle position alone may also be used in determining the switching point. Since, for a given flow rate, the metering valve will assume different positions dependent upon the pressure differential, the switching point will tend to be shifted with pressure. The benefit of this phenomenon is indicated in the main body of this report where a shift upward in flow with increasing pressure is shown for the intersection of the series and parallel efficiency curves at constant expander speed.

Whether or not the overshoots and undershoots that will want to occur during transition will be a significant factor will depend on the inertia of the expander and the response of the speed variator. An accumulator can mitigate these effects. It is also possible that the switching can operate on the variable speed drive to start the speed shift that the change in mode requires.

Deadband must be built into the switching mechanism to avoid oscillation at the switching point.

A hypothetical two-stage pumping system is shown schematically in Figure I-4, along with a simple 3-way pressure actuated selector valve.

The main high pressure pumping elements are shown as gear pumps. A centrifugal inducer charges either one or both high pressure stages, depending on the mode of operation.

The accumulator may be used to supply fluid for high flow transients and/or to dampen overshoots and undershoots during transition between the two modes of operation.

The check valve is to prevent back flow to the first element. In Figure I-2, the switching includes this function. All positive displacement pumps contain a relief valve for protection. In the Rankine cycle pump system, this valve may perform double-duty by also dumping flow at the high-speed, low-flow corner of the operating envelope.

The controlling parameters are not indicated in these figures. The selector valve reacts to a pressure signal which may be generated in many ways.

The benefits that accrue to the two-stage, series-parallel, pumping system are:

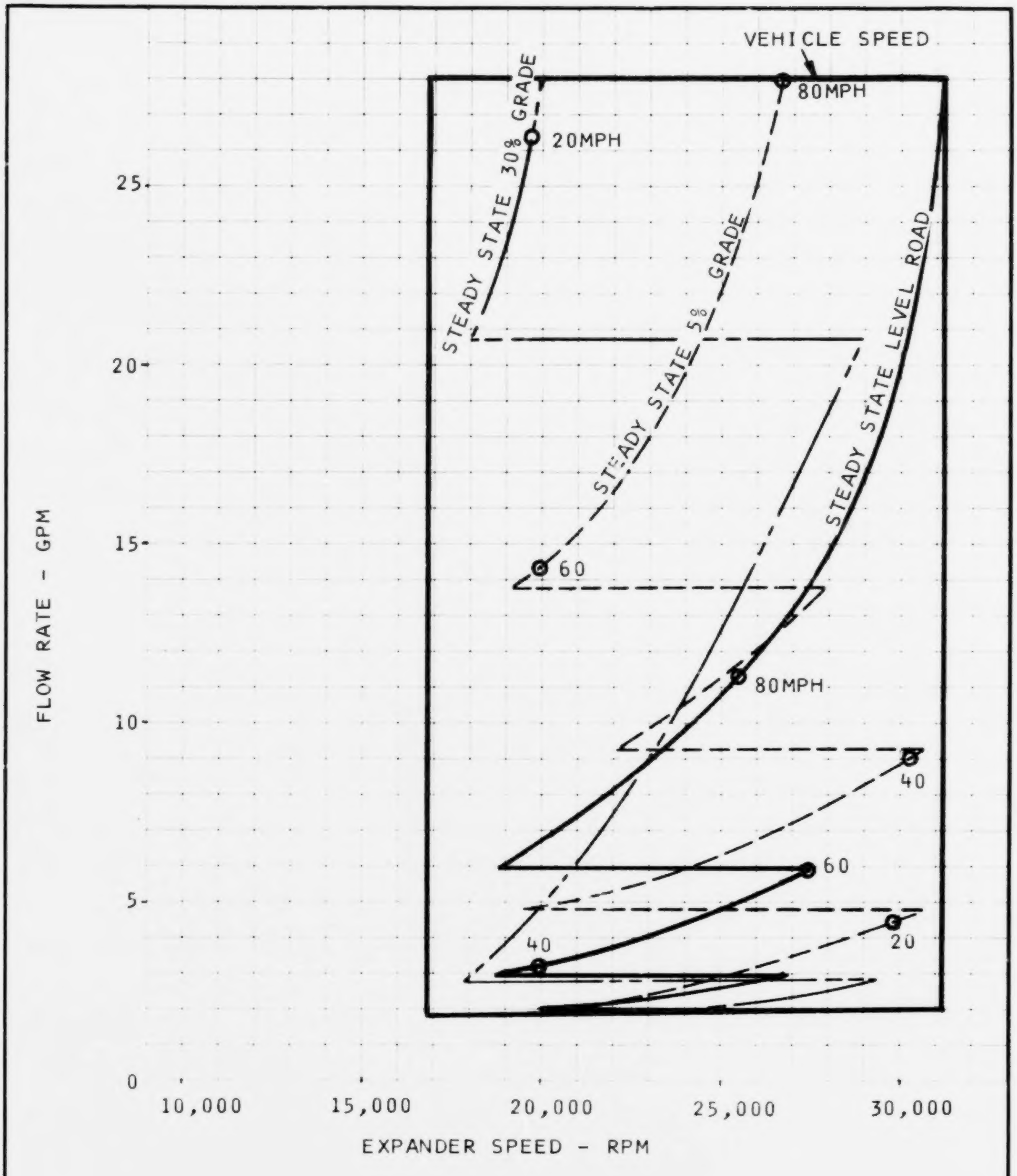
1. The efficiency can be maintained at higher levels over the full pump speed range.

2. The speed range of the variable speed drive can be narrowed, permitting more efficient operation of the drive.

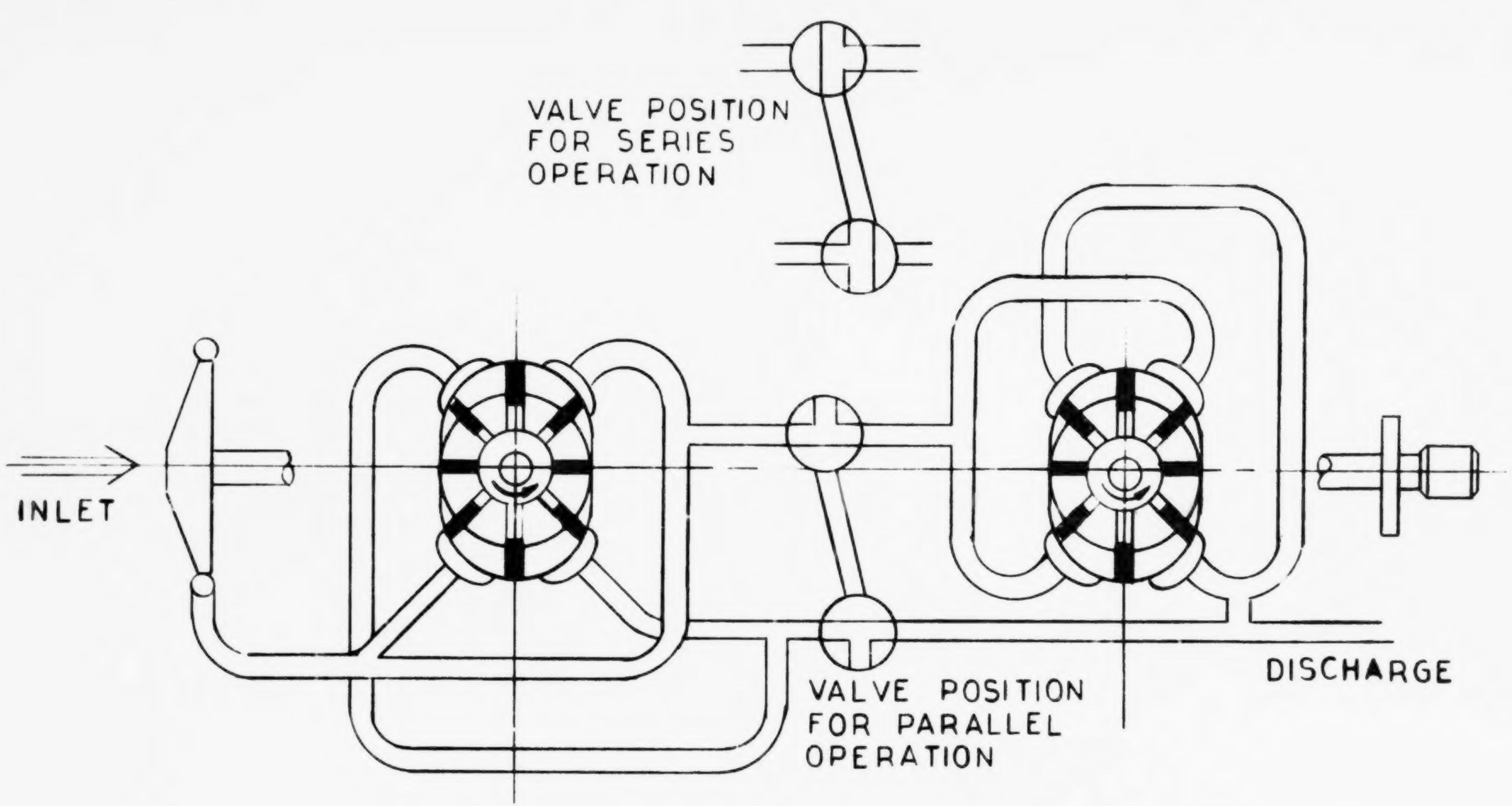
3. More of the engine operating envelope can be obtained without supplementary equipment.

The only apparent disadvantage is in the added system complexity. Furthermore, it remains to be seen whether or not discontinuities will occur during transitions. A more complete system analysis must be conducted to ascertain the measures to be taken to resolve this problem, if it exists.

Colt Industries Inc **Chandler Evans Inc Control Systems Division**



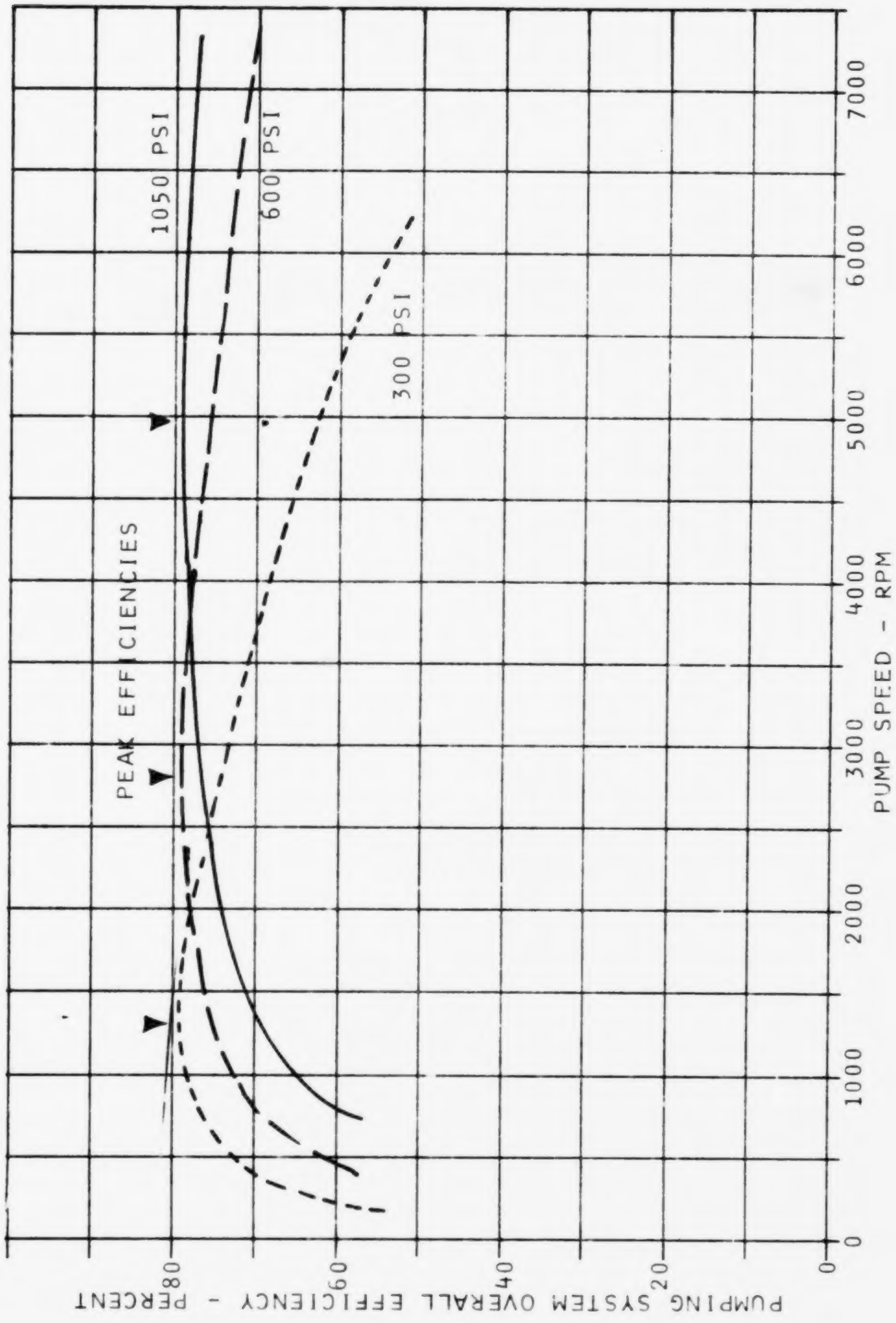
ENG. REPORT NO.	**REPORT TITLE**	**CURVE TITLE** FLOW REQUIREMENTS ORGANIC FLUID TURBINE ALRC
DATE		
PREPARED BY		**FIGURE NO.** I-1



TWO STAGE VANE PUMP

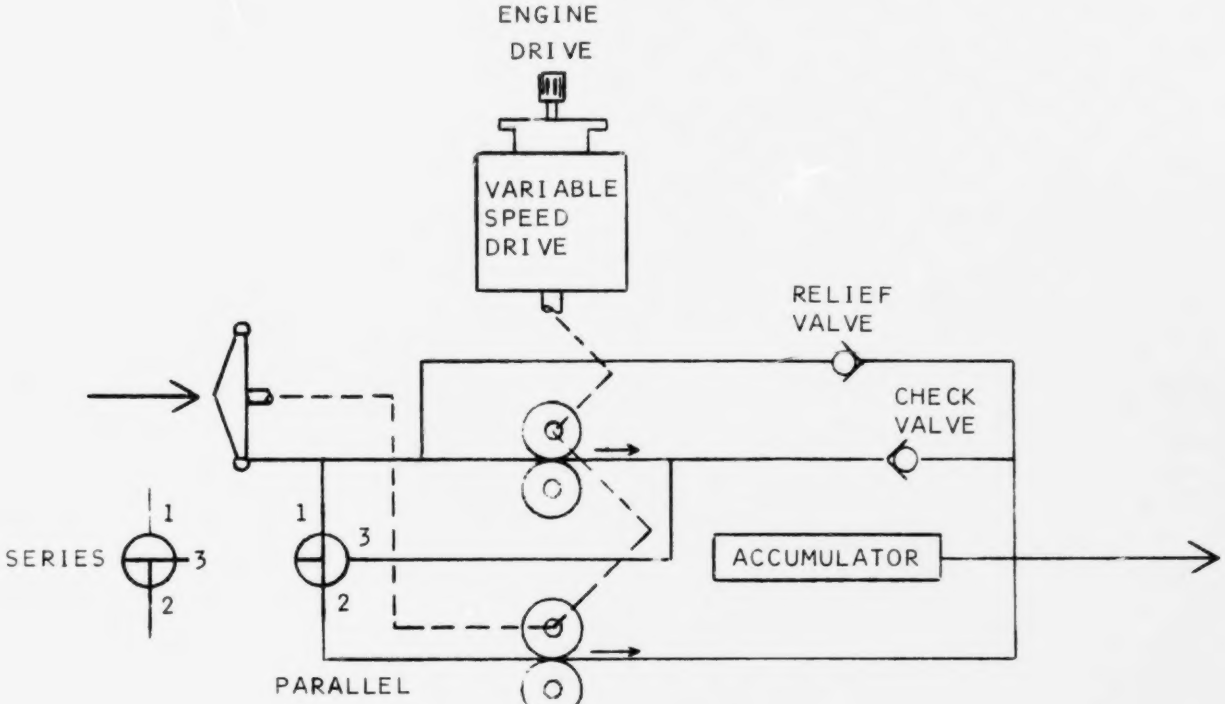
SERIES-PARALLEL OPERATION

FIGURE NO. I-2

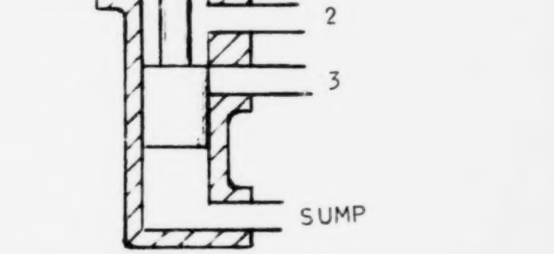


VANE PUMP EFFICIENCIES

FIGURE NO I-3



SCHEMATIC



SELECTOR VALVE

FIGURE I-4